GENERAL MOTORS

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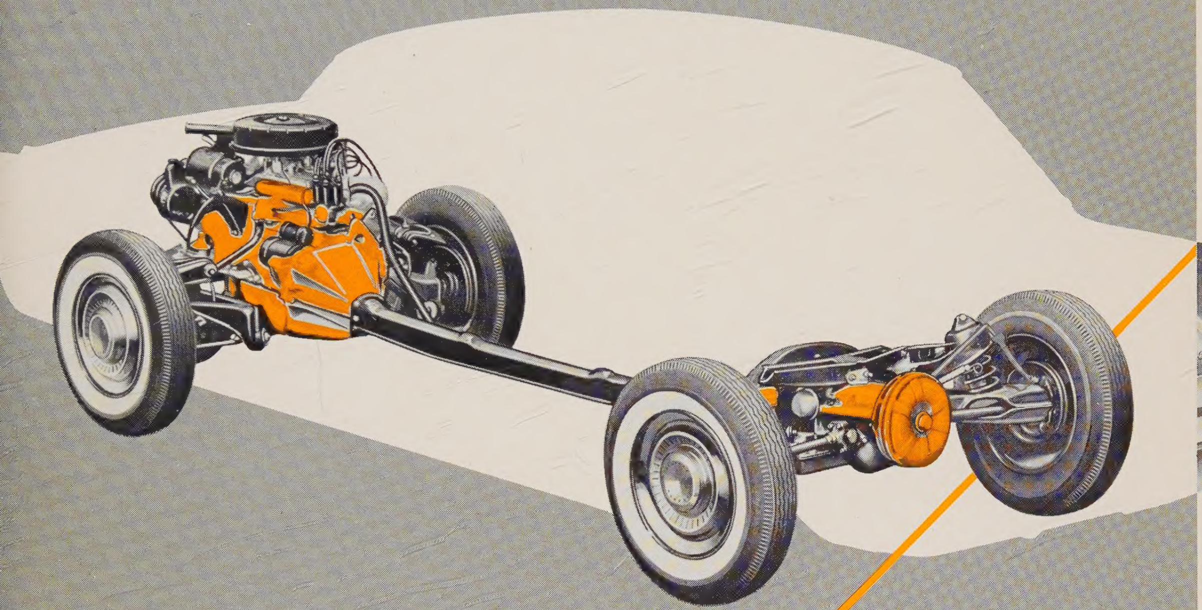
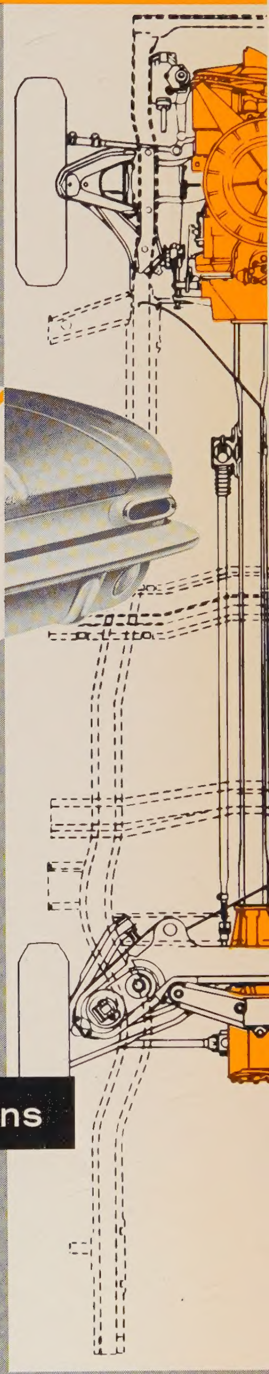
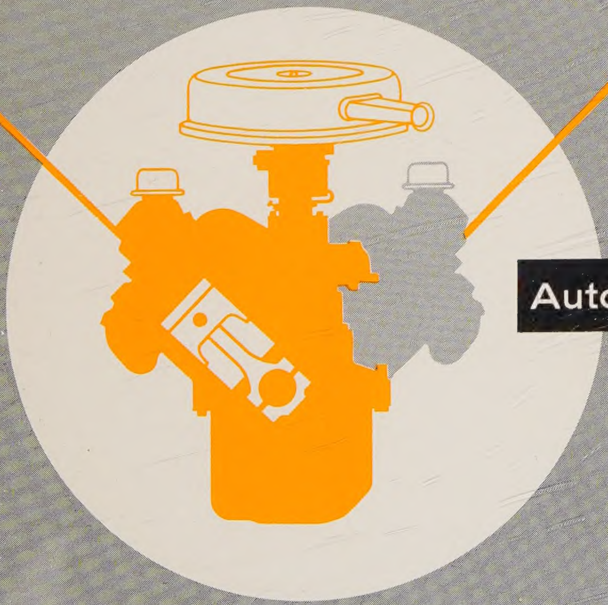
ENGINEERING JOURNAL

for educators in the fields of engineering and allied sciences







Automotive Design Innovations

On Writing and Speaking by Engineers

HIGH quality in design and accuracy in applying engineering fundamentals are, of course, primary requisites in an engineer's work. But every design, innovation, or new idea must be described or reported to someone else before it can take shape and perform a useful function.

To do this job effectively, the engineer needs to be proficient in the writing and speaking skills.

It seems hardly necessary to say this. Yet, educators report that many engineering students have only indifferent interest in their undergraduate courses in writing and speaking. Some students openly resent the "bother" of such courses. Some dislike being graded for English in certain technical courses. Other students believe that they actually won't need to do much writing or speaking in engineering work.

Such attitudes would be quickly dispelled if these students could look over the shoulders of most successful engineers today. They would see many occasions when the engineer needs his writing and speaking abilities. They would observe that the success or failure of a particular project or assignment often depends on the engineer's skill in expressing himself.

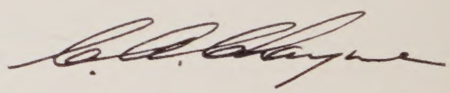
Examples of job-related writing or speaking assignments are numerous in industry. The following are just a few selected from General Motors engineering activities:

- Writing a report on a test of vehicle ride and handling at the GM Proving Grounds
- Writing a proposal for the purpose of obtaining management approval for a new engineering project
- Writing the technical portion of a proposal in connection with a formal bid on a Government defense project
- Speaking to an audience of people from several GM Divisions to exchange technical information
- Speaking to an audience to explain a design and promote its application.

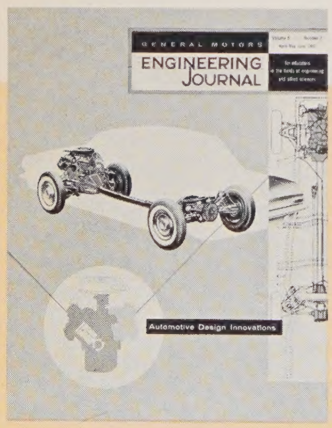
Besides these, there are other examples that are not a part of the engineer's regular job assignment, although the subject matter normally is related to it. He may write a paper for publication, or he may give a technical presentation at an engineering society meeting. He also may assist educators by speaking to appropriate student audiences both at the college and secondary school levels. The technical papers and listing of speaking appearances published regularly in the GENERAL MOTORS ENGINEERING JOURNAL are evidence of these types of assignments.

The student preparing for an engineering career can help himself by first recognizing that engineers do indeed use writing and speaking skills. Secondly, he can make use of every opportunity—both while in college and afterward—to practice and develop these skills. He should include in his educational program as much formal training in writing and speaking as possible. He also should apply the techniques, where appropriate, in all areas of his work. His laboratory report or term paper, for example, should have not only technical excellence but writing excellence as well.

The engineer who acquires a proficiency in expressing himself is best equipped for outstanding job performance and opportunities for advancement.



C. A. Chayne,
Vice President in Charge
of Engineering Staff



THE COVER

Design innovations have been characteristic of the United States automobile business throughout its long history. Easily remembered are ones such as independent front wheel suspension, the fully automatic transmission, and—more recently—the concept of the air cooled, lightweight, rear-mounted engine developed by Chevrolet for Corvair passenger cars and trucks.

On this issue's cover, artist Ernest W. Scanes recognizes another innovation in the U. S. automobile market—the concept of a front-mounted engine and a rear-mounted transmission connected by a flexible driveshaft. This was introduced on the 1961 Pontiac Tempest passenger car. Included in this concept is the adaptation of the right-hand bank of a regular Pontiac V-8 engine to produce a 4-cylinder, cast iron engine. This concept is one solution to the problems of weight distribution and handling which are more serious in the design of smaller cars. Two other GM smaller cars, the Oldsmobile F-85 and the Buick Special, use lightweight, aluminum V-8 engines and conventionally mounted transmissions.

CONTENTS

PAPERS IN THIS ISSUE

Page

CLASSROOM PROBLEMS

OTHER FEATURES

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THE GENERAL MOTORS ENGINEERING JOURNAL is published quarterly by the Educational Relations Section of the Public Relations Staff, General Motors Corporation. It is intended primarily as a medium for presenting to educators in the fields of engineering and allied sciences the results of General Motors research and engineering developments in both the product and production fields.

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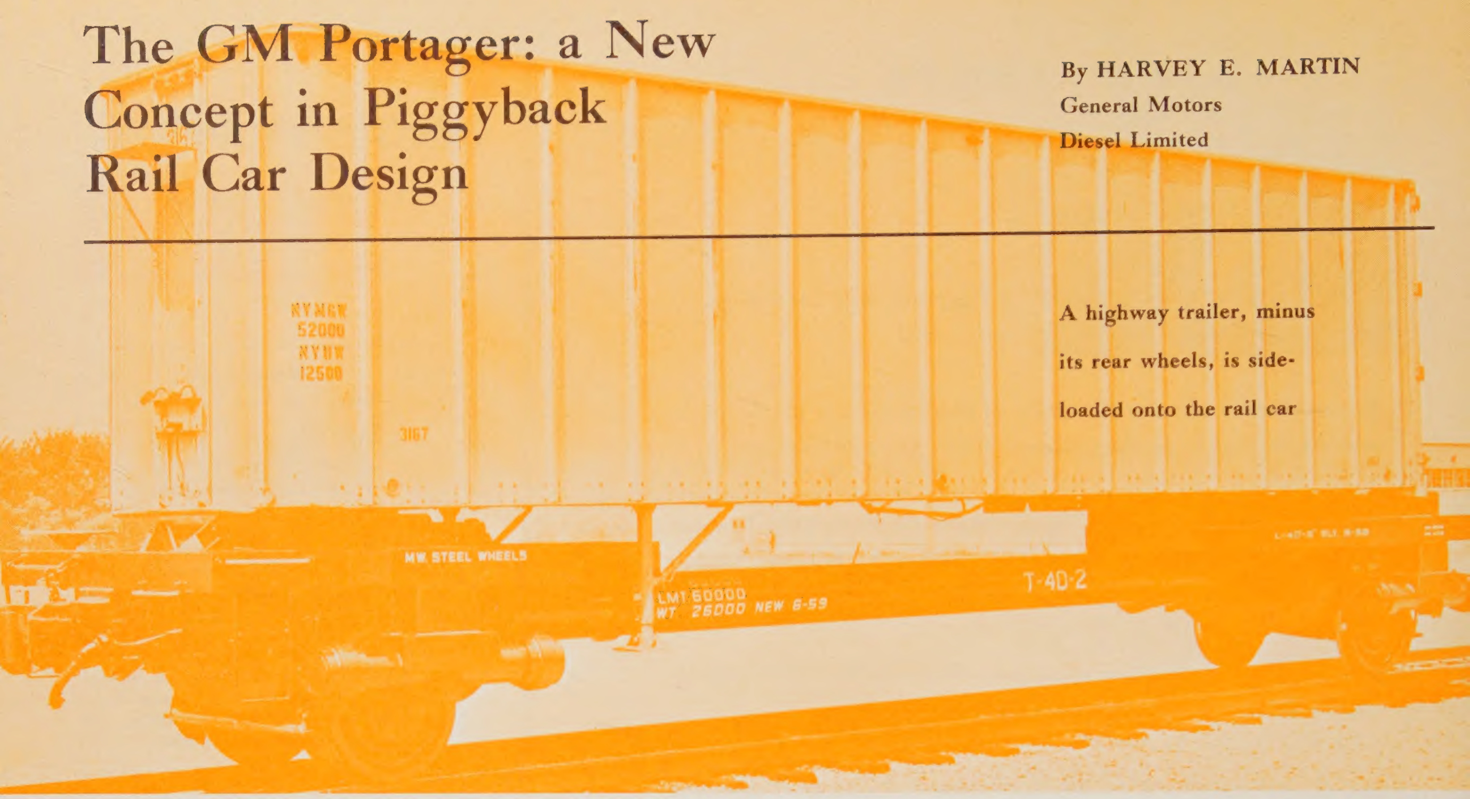
The GM Portager: a New Concept in Piggyback Rail Car Design

By HARVEY E. MARTIN
General Motors
Diesel Limited



A highway trailer, minus its rear wheels, is side-loaded onto the rail car

One of the newer developments in railroad operations is that of piggyback—the movement of highway truck trailers on specially designed railroad flatcars. The Canadian Pacific Railway Company, one of the largest piggyback operators on the North American Continent, recently conducted an investigation which showed that side-loading piggyback rail cars afforded certain advantages over the end-loading types currently in use. With a view toward integrating side-loading cars into existing equipment, the CPR gave the problem of developing a new concept in side-loading rail cars to General Motors Diesel Limited. Using their experience with car structures for Diesel-electric and Diesel-hydraulic locomotives, GM Diesel Ltd. engineers designed two test prototype cars to meet CPR requirements. These cars, which were named *GM Portager*, were tested for structural stability, impact resistance, and ride qualities. The information gained from these tests then was used to design and build two production prototype cars. These cars are now accumulating railroad mileage to evaluate reliability. The results obtained so far indicate that the unique design of this side-loading car can provide definite benefits to and improvements in piggyback rail operations.

Because of the dispersed population in Canada, transportation costs represent a higher percentage of the total service expense in the Canadian economy than in most other countries. This environment has resulted in a continual search for new methods and equipment within the Canadian transportation industry to increase efficiency and to reduce capital and operating costs.

One of the newer methods applied to Canadian transportation by its two major transcontinental railroads is piggyback—the movement of highway trailers on rail cars. The piggyback concept of railroading combines the speed and efficiency of the railroad in hauling high tonnages long distances with the economy and flexibility of the truck for terminal operations.

How a Developmental Program Started

One of the largest fleet of piggyback rail cars owned by an individual railroad belongs to the Canadian Pacific Railway Company. The current CPR piggyback system involves the use of flat cars having special adaptions for piggyback service. The system also requires special railway terminal facilities which consist of ramps and walkways to allow highway trailers to be end-loaded "circus fashion" onto the flatcars. Trains are broken into modules of from 10 to 15 cars for loading and unloading operations. A conventional highway tractor with an elevating fifth wheel hitch moves the trailer during loading and unloading. The deck of the flatcar has a king pin pedestal which is retracted during loading. After a trailer is positioned on the flatcar, the pedestal is raised to secure the king pin of the trailer during transit.

This system achieves the basic requirements of piggyback and produces satisfactory loading and unloading times. It was felt by the CPR, however, that the system had some shortcomings which perhaps could be improved upon.

The CPR investigated other piggyback systems and reached the conclusion that the side-loading piggyback rail car possessed certain inherent advantages over the end-loading type currently used. A side-loading piggyback rail car then was designed and built by the CPR from a conventional flatcar. Preliminary tests confirmed certain basic principles and indicated that the design of a new side-loading rail car system should be pursued.

At the request of the CPR, engineers of General Motors Diesel Limited, London, Ontario, were asked to design and develop a new system of side-loading for piggyback rail operations based on specific CPR requirements. A concentrated engineering developmental program was begun to expedite the new product through its several phases.

This paper represents a progress report on the developmental program—a program which has produced a new concept in side-loading, piggyback rail car design.

Why the Side-Loading Concept was Decided Upon

The concept of the side-loading piggyback rail car is based on several advantages which it offers to railroads and truck operators.

Trucking Operations Simplified

The current end-loading CPR piggyback rail car carries a highway trailer together with its rear wheels, or tandems. The proposed side-loading rail car would transport the trailer without its rear tandem, which would be left behind at the terminal. (A trailer transported in this manner is commonly referred to as a container). This would localize licensing, maintenance, and records. Also, the number of tandems required in a trucking operation would be reduced since none would be tied up in transit.

Wind Resistance Decreased

The absence of the trailer rear tandem would contribute to over-the-rail economy. A sizable portion of dead weight would be eliminated. This decrease in load would provide the rail car with a more favourable centre of gravity for stability and better tracking, bring the overall height of rail car plus trailer within all railroad system clearances, and would reduce train wind resistance. Since CPR piggyback trains run at passenger train speeds, a decrease in wind resistance would be a significant factor in reducing total train resistance and in helping to maintain schedules.

Rail Car Design Simplified

The rail car could be minimized—that is, aprons, guide rails, decking, and other special requirements to provide a path on the rail car for loading and unloading the trailer could be eliminated.

Faster Loading, Unloading Times

Side-loading would allow shorter loading and unloading cycles with fewer manual operations. This would improve terminal operation economy and provide faster delivery of goods to the customer.

Loading, Unloading Operations Simplified

Side-loading would permit the rail car to be loaded or unloaded from either side, from either direction, and in any sequence. This would eliminate certain directional and sequential loading restrictions which exist with the current system and permit faster handling of rush shipments.

Rail Car Design Had to Meet Specific Requirements

The design requirements established by GM Diesel Ltd. and the CPR were as follows:

- The first cost of the rail car should show a significant savings over existing piggyback equipment
- The rail car should be designed to require minimum maintenance
- The rail car must be capable of running satisfactorily at speeds up to 75 mph
- The vertical and lateral accelerations to which the highway trailer would be subjected while being transported by rail must be of the same order as those experienced in normal highway service. (This defined the ride requirements)
- Transmittal of railroad handling impacts to the trailer and to its load must be such that no damage to either will occur
- Highway trailers must require no special attachments or modifications that would significantly increase expense to the truck operator
- The rail car must be compatible with existing railway equipment to the extent that it must run in the same train
- The rail car should accommodate highway trailers from 35 to 40 ft long and be able to carry a load of 60,000 lb.

It should be noted that the restriction of standardization of components with existing conventional rail car equipment was not imposed on the design. This was contrary to the traditional approach and allowed the design to be radical, if necessary, to achieve the desired results.

The design of the side-loading piggyback rail car by GM Diesel Ltd. began with a large degree of freedom but with quite stringent requirements. As soon as primary layouts of the basic design concept were completed, ⅛-scale working models of a highway trailer, a proposed loader, and the rail car were built by GM Diesel Ltd. (Fig. 1).

The basic design concept established a rail car having a single centre sill member, single axle trucks, and a mounting for the trailer that utilized the king pin and rear guide angles of a highway trailer. Highway trailers with these guide angles, which provide for tandem shifting or removal, are coming into common usage. The sliding rear tandem, originally developed to allow truck operators to comply with different localized wheel base requirements, provides other advantages which add to the flexibility of a trucking operation. Also, axle loads can be equalized by tandem shifting.

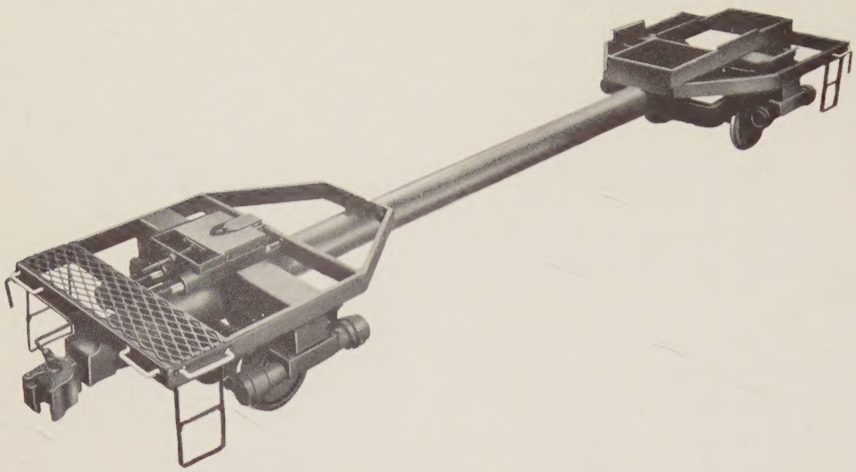
The loader proposed for use with the side-loading rail car had four-wheel steering and a loader table that would raise the highway trailer from underneath (Fig. 2). The open centre section of the rail car would allow the jack of the loader table to be positioned almost at the load centre. This would lessen the structural requirements for the loader table. Because the small diameter front wheels of the loader would pass under the centre sill, the need for having an over-hung counter balancing weight to give the loader stability when carrying a load was eliminated. The four-wheel steering feature of the loader also would provide for quick and accurate spotting of the load no matter how the trailer was aligned relative to the rail car (Fig. 3). It should be noted that the rail car design put little restriction on the methods or equipment used for side-loading. Any method using equipment that would perform the lifting and transfer operations without interference or damage to the trailer could be utilized.

Test Prototype Cars Built

When the preliminary design was complete, two test prototype cars were built and then subjected to a series of developmental tests to evaluate the unique design features. For identification, the test prototype cars were given a model designation T-40 (Transporter for 40-ft. trailers). The rail car was later named *GM Portager*. Included in the unique design features of the GM Portager are the underframe, truck suspension, and trailer mounting (Fig. 4).

Underframe

A 16-in. diameter, single tubular centre sill was selected to give high buff and drag



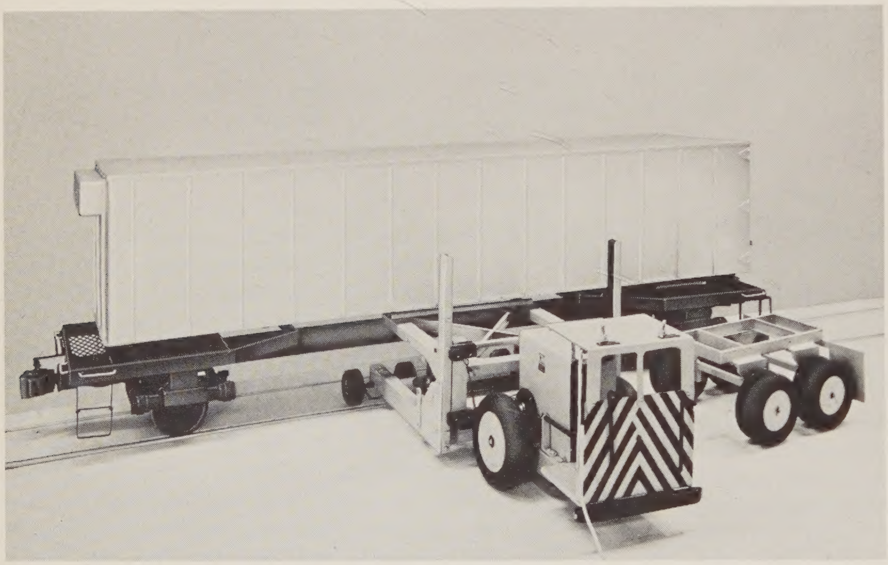


Fig. 1—As soon as the basic design concept of the side-loading piggyback rail car was established, the ⅛-scale working models shown here of the rail car, highway trailer, and proposed loader were built. These models were a significant visual aid in developing design improvements, overcoming loading and unloading restrictions, and demonstrating the design of the new piggyback system.

strength and relatively low torsional resistance to assist in the track holding ability of the car under adverse conditions. No significant beam strength to support the vertical load of the highway trailer was required. The vertical load is directly supported above the trucks at each end of the rail car. Rubber draft gears were used with standard Type E couplers at each end of the centre sill member which, based on locomotive experience, had sufficient capacity to protect the GM Portager from railroad impacts. Air brake piping was routed through the centre sill tube. Other details of the underframe involved bolster structure for the truck suspension, container mounting, air brake equipment mounting, and the attachment of various other minor but essential equipment, such as the uncoupling rods and the handbrake.

Truck Suspension

A two-wheeled design was selected for the GM Portager because the maximum load of 60,000 lb was within the load carrying capacity of four wheels per rail car, using 33-in. diameter wheels. A swing hanger suspension was chosen to give easy lateral action—a requirement for high speed operation (Fig. 5). A swing hanger length of 10 in. was arbitrarily selected. To develop a high degree of roll stability, a four-point swing link

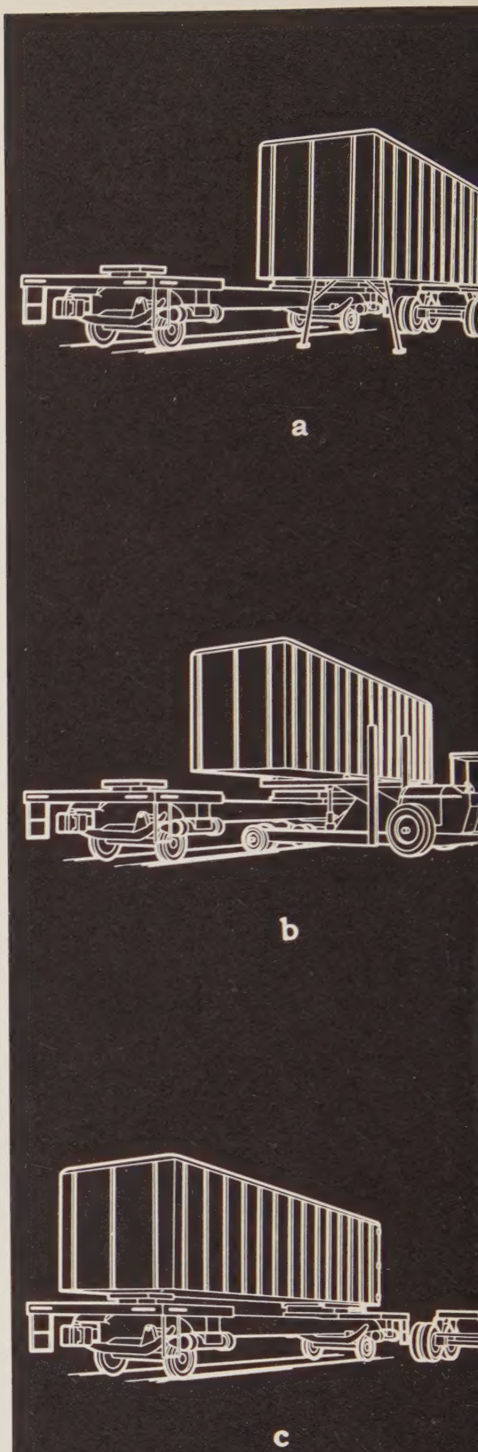
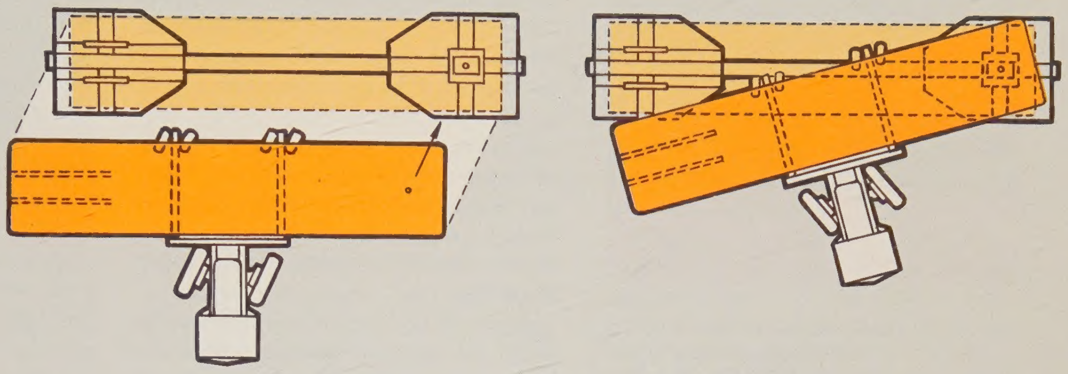


Fig. 2—Shown here are three sequential positions of the piggyback system equipment during a loading operation. In (a), the highway trailer together with its rear wheels, or tandem, is parked and is ready to be side-loaded onto the rail car. In (b), the loader has manoeuvred into position under the trailer, lifted it off its rear tandem, and is in transit with the load toward the rail car. In (c), the trailer has been positioned and lowered onto the rail car. The king pin hitch of the rail car is secured to the king pin of the trailer. Unloading is accomplished by a similar procedure in reverse sequence.

. 3—Quick and accurate spotting a highway trailer can be achieved ough four-wheel steering of the der. Besides providing a short turn- radius, four-wheel steering makes sible two loading manoeuvers which ninate the need for precise spotting parallel locating requirements. st, by keeping the wheels of the der turned out parallel with each er, the loader can approach the rail at any angle and aim the king pin the trailer at the hitch on the rail (left). In this case, the loader gresses sideways while moving for- rd. Second, with the trailer king pin d in position over the hitch on the car, the trailer can be rotated about centre of the king pin until the r guide angles of the trailer are in e with the mounting angles on the car (right). A trailer that is parked of line and at an angle to the rail , therefore, can be spotted on the car in one continuous motion using combination of these two loading noeuvers.



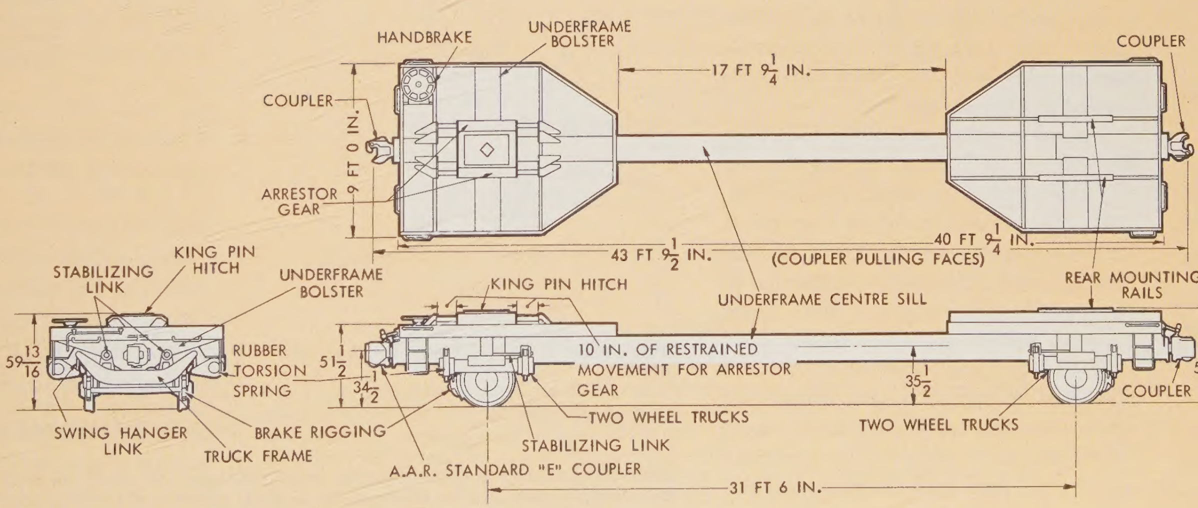
attachment to the underframe was used. The geometry for this arrangement was determined from calculations based on equating the resistive force at the wheel flange against the rail to the maximum simultaneous overturning force that could be developed on a curve by a loaded car (Table I).

The four swing links are connected to the arms of torsion springs mounted on the underframe. The flexible torsion-resisting elements of each spring consist of two pre-loaded rubber bushings. This spring design was chosen to provide a simple swing link connection, to act as a structural member, and to give a degree of vertical impact dampening. The idea of using rubber as a torsion spring element was conceived when, in an attempt to remove metal wearing bushings from a previously designed metal rod torsion spring design, it was found that the rubber bushings loaded in shear would provide most of the required torque reaction.

One of the test prototype GM Portagers had the axle fixed and each wheel rotated on tapered roller bearings mounted in the wheel hub. This was done to assess any benefit that the differential action of free wheels might give to a four-wheel rail car on a curve. The second test prototype car had wheels pressed on a rotating axle and inboard roller bearings.

The truck frame was fabricated from two longitudinal box sections welded to a

. 4—Shown here is the eral arrangement of the 1 Portager prototype rail fabricated for develop- ntal test purposes. This e-loading, two-axle, piggy- ck rail car is designed to ry highway trailers varying length from 35 to 40 ft. The o-wheel trucks are sus- ded at four points from the derframe by a swing hanger k arrangement. Truck inging is provided by four bber torsion springs. The derframe is composed of a gle centre sill member, ck support bolster mem- s, trailer support, and ac- sory structure for mounting highway trailer.



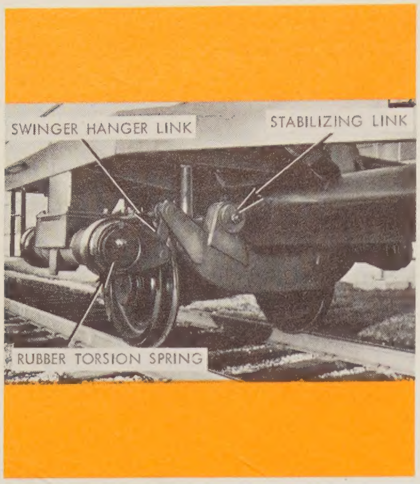


Fig. 5—The truck suspension features (a) a swinger hanger suspension to give easy lateral action during high speed operation, (b) a four-point swing link attachment to the underframe to provide a high degree of roll stability, and (c) rubber-bushed stabilizing links to transmit braking and acceleration reactions to the underframe. Each swing hanger link is connected to an arm of the rubber torsion spring.

traverse flamecut member at each end. Other features of the truck suspension include rubber-bushed stabilizing links to transmit braking and acceleration reactions to the underframe, automotive-type diaphragm single-acting brake cylinders, and rubber block bolster lateral stops mounted on an angle. The lateral stops also act as cushion seats in the event of failure of any vertical suspension member.

Trailer Mounting

In order for the GM Portager to carry standard removable-tandem trailers, the trailer mountings on top of the underframe were designed to duplicate the manner in which the trailer would be carried in highway service.

Support for the trailer is provided at the rear of the Portager by rails having the same spacing as tractor tandem support rails. At the front of the Portager, a highway tractor king pin hitch, or fifth wheel, is used. This hitch is connected to the underframe through a dual, high-capacity rubber element arrestor gear (Fig. 6). This gear allows up to 10 in. of restrained movement of the trailer fore and aft from the centreline of the hitch. This movement cushions the container and its load from railroad impacts. Frictional forces at the arrestor gear and rear rails add dampening capacity and prevent continual fore-and-aft jostling of the trailer while in transit.

Underframe Structural Tests

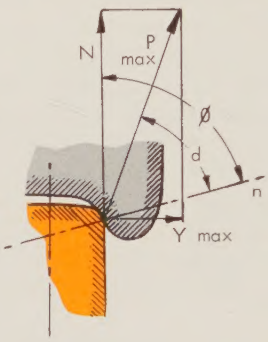
Before the test prototype cars were assembled, the underframe was structurally tested. Dynamic values of all loads were applied separately and in combination to the underframe. Stress levels at all critical locations then were measured. The results were analyzed graphically for fatigue using empirical formulae proven by experience to give reproducible and predictable results.

The structural tests were performed by mounting the underframe on a very rigid test fixture and applying the dynamic loads by means of hydraulic jacks. The fixture used for the tests weighed over 70,000 lb.

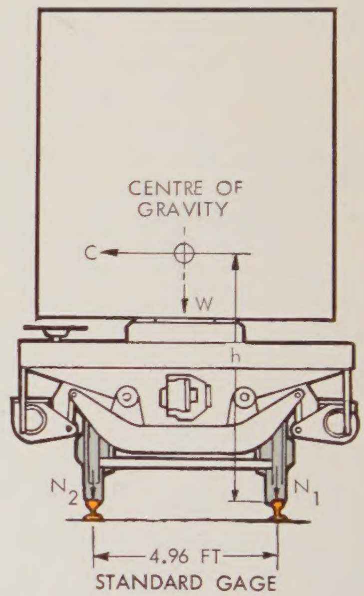
Areas of high stress were generally known from design calculations. To locate stress concentrations precisely and to make certain none were missed, the brittle lacquer technique for determining stress was used. A brittle lacquer coating was sprayed on the structure and the loads then applied. The line patterns formed when the lacquer cracked indicated both the general location and direction of strain. Since the sensitivity of the brittle lacquer had been calibrated, the approximate stress levels were known.

Direct reading, resistance wire strain gages were applied to areas where the brittle lacquer line patterns indicated strain. Stress values then were calculated for all desired load combinations. For the underframe test, 185 strain gages were used. Vertical, buff, impact at arrestor, and twist loads were applied separately and in desired combinations.

The test results indicated one area of



WHEEL-TO-RAIL FORCES
(NEW WHEEL — AVERAGE RAIL)



CENTRIFUGAL FORCE:

$$C = \frac{(W)(V^2)}{(g)(R)}$$

WHEN $N_1 = 0$ (POINT OF OVERTURNING):

$$Y = \frac{C}{2} = \left(\frac{W}{2}\right)\left(\frac{4.96}{h}\right)$$

WHEN WHEEL IS AT THE POINT OF CLIMBING THE RAIL:

$$N_2 = (Y)\cot(Ø - d)$$

C = CENTRIFUGAL FORCE (LB)

h = HEIGHT FROM TOP OF RAIL TO CENTRE OF GRAVITY

g = ACCELERATION DUE TO GRAVITY (FT PER SEC^2)

V = VEHICULAR SPEED (FT PER SEC)

R = RADIUS OF CURVE (FT)

W = TOTAL WEIGHT OF VEHICLE AND LOAD (LB)

Y = LATERAL FORCE AT EACH WHEEL (LB)

N_1 AND N_2 = WHEEL LOADS (LB)

P(MAX) = RESULTANT REACTION OF RAIL LOADS (LB)

Table I—The diagrams above show the loaded vehicular and wheel rail forces present when a loaded rail vehicle negotiates a flat curve or, for practical purposes, a super-elevated curve at a speed above the balance point. The equations shown were used in calculations to balance the overturning and rail climbing tendencies of the GM Portager. The calculations showed that, while traveling at normal railway operating speeds for any given curve, the GM Portager would not overturn while carrying a full, high centre of gravity load. The calculations also showed that as the load is decreased, the overturning tendency is decreased.

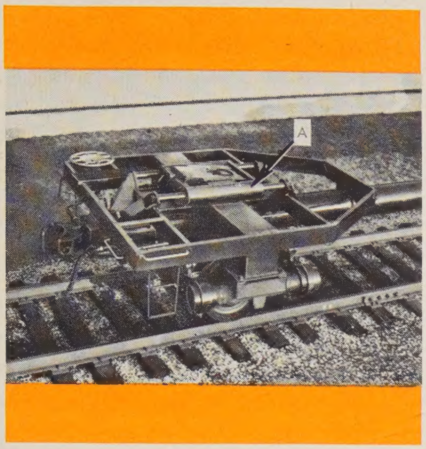


Fig. 6—A conventional highway tractor king pin hitch, or fifth wheel, is used at the front of the GM Portager to help secure the trailer to the rail car. The hitch is connected to the underframe through two rubber element arrestor gears *A*. The gears allow the trailer to move as much as 10 in. fore and aft from the centreline of the hitch. Such movement acts to cushion the trailer and its load from railroad impacts.

the underframe which required a major design change due to overstress and also some minor revisions to eliminate localized stresses. The underframe bolster section—that is, the box section that supports the outboard torsion springs—was overstressed in bending. This section was rebuilt and retested until the stress level was brought within acceptable limits (Fig. 7).

The twisting test indicated an underframe flexibility that, for a fully loaded car, would allow one axle to be over two degrees out of parallel with the other axle with negligible underframe stress. It was felt that this flexibility would make a marked contribution to the rail car's ability to hold the rail even under adverse conditions.

The frame for the truck also was structurally tested in a manner similar to the underframe test. Fifty strain gages were used for the truck frame test. No overstressed areas were found, although minor modifications were made to reduce two stress concentrations.

Developmental Tests Conducted Jointly with the CPR

The developmental test program for the GM Portager consisted of performing over 300 different instrumented tests on four different railroad locations and under a number of loading variations and numerous running speeds up to 75 mph. Also, tractor-trailer test runs were made on local highways with the same trailer and load used in the railroad developmental tests. Additional tests and demonstrations were made which involved only observation, such as loader evaluation and adverse buff conditions. Over 11,000 test miles were accumulated by one of the prototype GM Portagers during the developmental test program.

All developmental tests were prescribed and conducted jointly with personnel of the CPR's Mechanical Engineering, Piggyback, and Research Departments. The CPR also provided a caboose in which was installed power supply, test instrumentation, communication equipment, and signalling devices (Fig. 8).

The test instrumentation installed in the caboose was selected to record speed, suspension and trailer movement, ride accelerations, and significant stresses and loadings. Three four-channel ride recorders were used to give a record of vertical, lateral, and longitudinal accelerations*. Motions, stresses, and loads were plotted by a multi-channel recording oscillograph. A calibrated recording tachometer provided accurate speed records. Signalling pens on these recorders provided synchronization as well as a running record of such road conditions as curves, crossings, switches, and bridges. Signalling keys were operated by an observer stationed in the cupola of the caboose. Periodic observations and measurements provided records on wheel wear, torsion spring angular location, trailer load behaviour, and other similar noteworthy conditions.

A base line for ride requirements was established from test runs on various local highways. The same trailer body and test load as used on the railways was used for the highway test runs. Preliminary comparative records were established from railroad test runs and impact tests made on a conventional CPR piggyback loaded rail car. This CPR car was included in the railroad test train along with the prototype GM Portager on a number of test runs for further direct comparisons.

*See the *Appendix* to this paper on page 10 for a discussion of typical records obtained from the ride recorders.

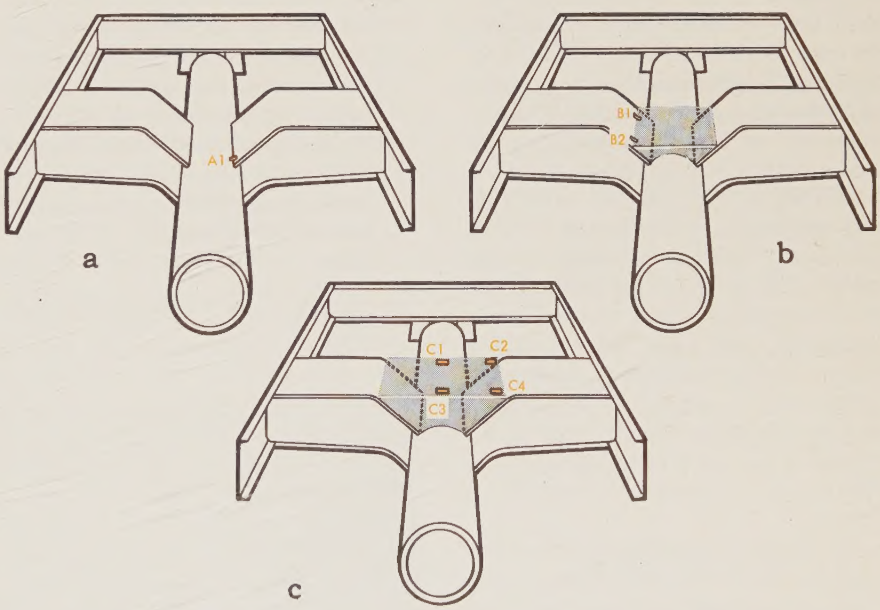


Fig. 7—Structural tests performed on the underframe showed that the underframe bolster was overstressed in bending where it joined the top of the tubular centre sill. When the structural tests were first performed, strain gage *A1* recorded 50,000 psi at 200 per cent loading (a). The box structure shown in (b) then was added. This was the largest structure that could be added and still provide space between the two bolster arms for the trailer hitch arrestor gear. This redesigned section reduced the maximum stress level, as indicated by strain gages *B1* and *B2*, to 38,000 psi. This was still above desired limits. It became necessary, therefore, to add the reinforcing structure shown in (c) and to redesign the arrestor gear mounting arrangement. This design change produced a maximum stress level well within limits. Strain gages in *C1*, *C2*, *C3*, and *C4* all indicated a stress level below 7,000 psi. The high stresses were principally caused by the notch effect of the bolster-to-sill joint in the first two designs (a) and (b). The third design (c) eliminated the notch effect and the high stresses as well.

Evaluation of the test prototype GM Portager was started on a local branch line where light-weight rails and sharp curves were available. This branch line also provided the opportunity to aggravate the track conditions artificially by temporarily displacing the rails. The track holding ability and general road worthiness of the Portager were evaluated on this branch line. Then, higher speed runs were carried out on a section of main line where curves were sharper and more numerous than normal. Final developmental tests were conducted in revenue service in scheduled Toronto-to-Montreal CPR piggyback trains.

In conjunction with these tests in revenue service, two prototype loaders were tried out. The results showed very encouraging times for loading and unloading.

The loaders were developed and built for the CPR by a United States manufacturer after consultation with GM Diesel Ltd. engineers. The loaders were powered by standard highway tractors and provided an additional feature of moving the complete Portager from one track to another by lifting under the centre sill.

Included in the developmental program were two series of impact tests made on track located within GM Diesel Ltd. property. The first series of tests consisted of impacting a conventional CPR piggyback rail car carrying a loaded trailer against several box cars which had their brakes set. At 7½ mph, the trailer king pin was pushed back about six in. with consequent structural damage to the trailer. This series of tests indicated the need for accurate speed control, instrumentation, and other aids to record stresses, deflections, relative movements, accelerations, and speeds. This was necessary to allow the impact speed to be gradually increased up to the point of damage and to provide a record of the results.

The second series of impact tests, which represented an improved procedure, was made with a loaded prototype GM Portager. The trailer was loaded with steel shot in 100 lb bags to a gross weight of 49,000 lb. A loaded coal car was coasted down an incline to impact against the Portager. Impact data were recorded on a multi-channel, high speed photographic type oscillograph. Also, high speed movies were taken. Motion, stress, and speed transducers were engineered by GM Diesel Ltd. Instrumentation, high speed movie camera, and technical assistance were provided by the National Research Council of Canada. Impact speeds were gradually increased up to eight mph. At this speed of impact, the king pin hitch failed.

Various Modifications Made on Prototype Rail Car

Throughout the developmental test program, various modifications were made on the test prototype rail cars and then evaluated. The modifications made were based on test results which indicated their necessity if design requirements were to be achieved. Noteworthy among the modifications were:

- A lengthening of the truck suspension swing hanger links to reduce both the frequency and restoring force in order to slow down the lateral accelerations imposed on the trailer and its load
- A reduction in the spring rate of the torsion springs to reduce vertical accelerations
- Provision for secondary springing between the underframe of the rail car and the trailer mounting to reduce further the vertical accelerations. This provision was necessary because close limits on the coupler heights did not allow enough primary spring deflection to achieve a sufficiently soft vertical ride for the trailer
- Application of a different king pin hitch.

The truck suspension proved to be very stable throughout the test program. Roll was negligible, even after the spring modifications and with a heavy, high centre of gravity load in the trailer. The modified truck suspensions achieved a vertical ride equivalent to that obtained on the average highway. Excellent track holding ability at high speed was obtained from the primary suspension, which absorbed track irregularities without producing any tendency to sustained oscillation and vibration. The test prototype car with the freely rotating wheels on a fixed axle showed no apparent advantage in curve negotiation over the conventional rotating axle arrangement on the other test prototype.

The Portager ran straight and true over the full range of load and speed conditions. Derailment did not take place even under aggravated test conditions, such as one test which involved pushing an unloaded Portager on an 11½° curve with the full available starting tractive effort of four Diesel-electric locomotives.

The combined shock absorption capacity of the draft gear and king pin hitch arrestor gear proved capable of protecting the Portager, the trailer, and the load at impacts up to eight mph.

Production Prototypes Built

The experience accumulated during the developmental test program has been used in the design and building of two production prototype GM Portagers. Standard wheels and axles with outboard bearings and a softer suspension are the most significant design changes between the test prototype and production proto-

Fig. 8—Developmental tests on the two test prototype GM Portagers were conducted jointly with the Canadian Pacific Railway Company. The CPR provided a caboose to house instrumentation and related test equipment. This caboose *A* is shown positioned between an unloaded Portager *B* and a loaded Portager *C* while running on one of the four different railroad locations used during the developmental test program.

type rail cars. The production prototypes, in addition, represent a design having reduced frame stresses and reduced manufacturing costs.

Structural tests made on production prototype Portagers have confirmed the increased strength and lower stress levels in the underframe. Changes made in the two-wheel trucks have had no measurable detrimental effect on the track holding ability. An improved vertical ride has been achieved as a result of the suspension modification.

A number of additional tests, mostly prescribed by the CPR, were made on the production prototypes with satisfactory results. The most severe test was that made to assess the derailing tendency of an unloaded Portager when placed at the head of a 3,500-ton, 55-car freight train pulled by four Diesel locomotives. Repeated pull tests and train run-in tests were made with this train on a railroad line having grades up to two per cent and curves up to nine degrees. Run-ins were produced by applying maximum dynamic braking on the four Diesel locomotives with a stretched train. This produced an instantaneous head-end retarding force of 160,000 to 170,000 lb at speeds from 20 to 25 mph. Extremely severe impacts occurred due to the run-in of the train. Results of this test relieved concern of possible derailment due to positioning of unloaded GM Portagers at the head end of trains.

Signal tests also were performed. These tests showed that an unloaded GM Portager would satisfactorily operate the three types of railroad signalling equipment—a-c, d-c, and coded (d-c pulse).

Two noteworthy measurements were taken during tests on the production prototypes. The first showed that maximum shocks due to run-in and run-out in high speed freight trains were in the order of 2½ mph impacts. The second measurement showed that the vertical impact dampening of the primary rubber torsion springs reduced the most severe road shocks by a factor of eight.

Further testing was performed on the production prototype GM Portager by impacting the Portager against several box cars as done in earlier tests. Gross trailer loads of 60,000 lb were used. From these impact tests it was evident that the maximum impact resistance speed depends upon a number of variables such as type of load, method of packing, weight of load, trailer construction, and the nature

Fig. 9—A loaded GM Portager (left) has less dead weight and offers less wind resistance than a conventional piggyback rail car carrying a highway trailer with an identical load (right). The lower profile of the GM Portager brings the over-all height within all railroad system clearances. Also, the lower profile permits the GM Portager to use passenger train stations and sheds.

of the striking vehicle. With current arresting gear, little damage to GM Portager loads should be anticipated below eight mph impacts. This is at least equivalent to current equipment on which no damage is being sustained in regular piggyback service. The GM Portager is structurally capable of withstanding impacts of 12 mph.

Summary

This paper has described the design, development, and testing of a unique piggyback rail car and a related loading and unloading system to the completion of the production prototype stage. The over-all developmental program, which included a number of design changes to improve standardization, reliability, and adaptability, has proven both the rail car and the system to be sound.

The economies of complete GM Portager piggyback trains with current average revenue loading are now apparent, although not specifically determined. Less horsepower will be required to pull the Portager piggyback train because the average loaded weight is reduced by approximately 20 per cent, the lower profile provides less wind resistance (Fig. 9), and there are half as many wheels and bearings. Reductions in loading and unloading man hours of over 50 per cent, as compared with circus loading, have been demonstrated. A potential exists for further development in this area. Potential savings also exist in the areas of rolling stock investment, trailer tandem investment, maintenance (due to fewer wearing parts), stable high speed tracking, soft riding, and impact protection features.

Developmental work on the GM Portager still continues. The results achieved to date, however, show that the GM Portager has proven to be a significant step forward in piggyback train operations and can provide resultant benefits to railroads, truck operators, and shippers alike.

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Acknowledgement

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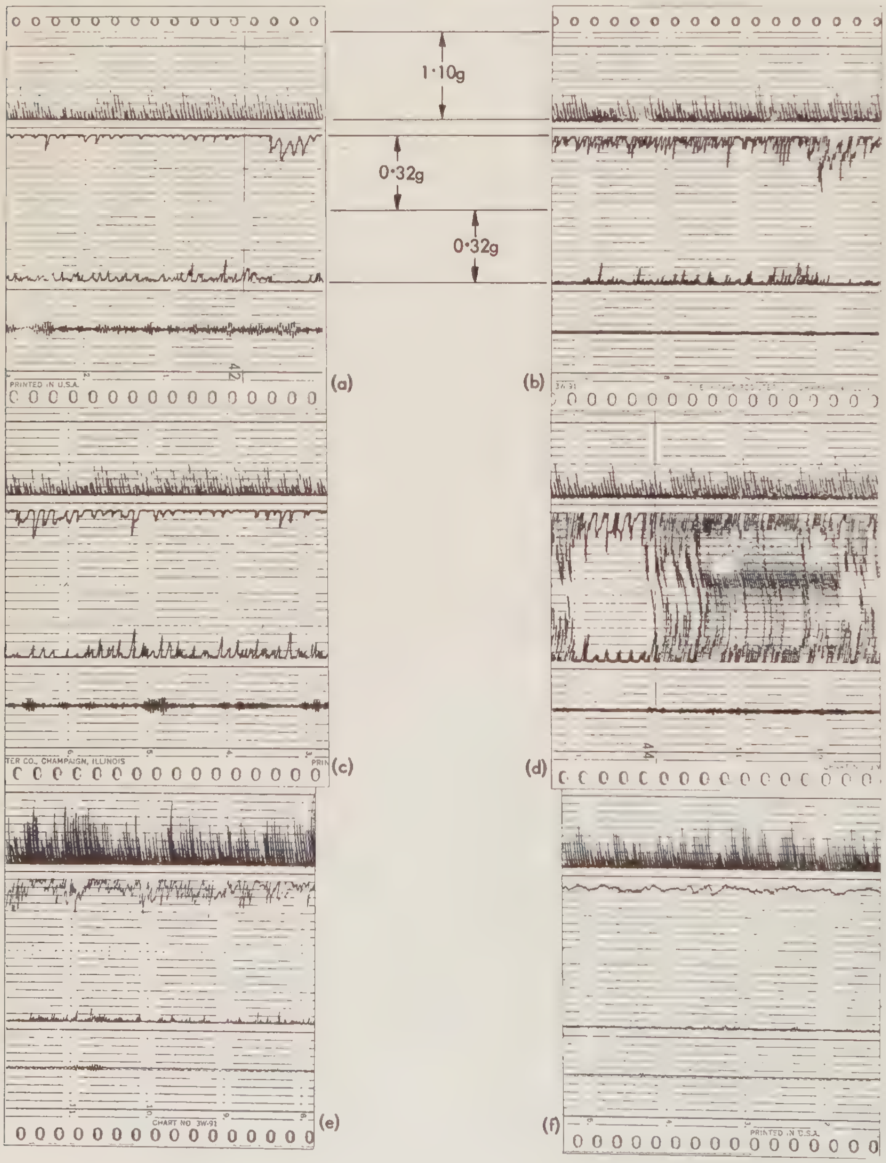
During the development of the GM Portager piggyback rail car, a record of the vertical, lateral, and longitudinal accelerations was made by means of four-channel ride recorders. Comparison data were taken for the performance of the Portager and the conventional CPR piggyback rail car. In all cases, the ride recorder was placed in the centre of the floor of the highway trailer, and the trailer loads were the same.

The charts shown in this Appendix are typical of those produced during the developmental program. The top trace on each chart shows vertical accelerations, the second trace shows lateral accelerations to the right, the third trace shows lateral accelerations to the left, and the bottom trace shows longitudinal accelerations. The vertical and lateral accelerations have the same calibration in g's for all charts shown, as indicated in charts (a) and (b). The bottom trace showing longitudinal accelerations should be ignored in the discussion to follow, since the calibration was not the same for the charts shown here.

Charts (a) and (b) illustrate the comparative ride of the GM Portager (a) and the CPR piggyback rail car (b) running at a speed of 50 mph over the same section of mainline track (mileage 16—CPR Belleville subdivision). Note the same railroad curve on each chart where each car leans to the right. Charts (c) and (d) illustrate the ride action of the GM Portager (c) and the CPR car (d) running at a speed of 65 mph on the same section of track (mileage 18—CPR Belleville subdivision).



A comparison between charts (b) and (d) shows peculiar lateral accelerations of the CPR piggyback rail car at the higher speed. This was typical of observations made on this car and resulted from rapid "nosing" of its four-wheel, box car type truck. The nosing action is caused by the rotation of the truck about the vertical axis of the centre bearing. As the vehicle speed reaches approximately 60 mph, the nosing becomes self-sustaining on straight track and is interrupted only by some positive force on the car, such as the centrifugal force encountered while rounding a curve. The effect of a curve can be seen on chart (d). The truck used on the GM Portager, although subject to the same force, does not change its lateral action to any significant degree as the speed increases. This is illustrated by a comparison of charts (a) and (c) for speeds of 50 mph and 65 mph, respectively. One of the main reasons for this is the easy lateral restraint of the truck suspension.

Charts (e) and (f) illustrate the comparative ride of a highway trailer having a conventional rear tandem when hauled by a tractor at 50 mph over two representative sections of highway. Chart (e) was produced for a section of highway No. 3—a two-lane highway with asphalt repair patches—near Blenheim, Ontario. Chart (f) was produced for a section of highway 401—a new, concrete four-lane highway—near Leamington, Ontario. The lateral accelerations shown in these charts are all to the right because of the crown of the highway.

The comparison charts provided information from which certain conclusions were drawn. First, the vertical ride of the GM Portager is equivalent to the CPR piggyback rail car. The CPR car has a soft primary suspension and, since it carries the rear tandem of a trailer, also has the tandem springing and rubber tires to act as a secondary suspension at the rear. Second, the vertical ride of the GM Portager is at least equivalent to that obtained with highway hauling. This is probably the most important measure of the effect of piggyback rail operation on the life of a trailer. Finally, the lateral action of the GM Portager is no more severe than that encountered in highway hauling and, as the magnitude of the lateral action indicates, is a minor factor in shortening the life of the trailer.

A Discussion on the Out-of-Roundness of Machined Parts and its Measurement

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Surveying the present state of out-of-roundness measuring techniques

Making many diameter measurements of a machined part is not sufficient to determine if the part is round. In the past, the determination of whether a part was round or out-of-round was made by inspecting the part in a V-block in conjunction with a suitable indicator. Recent demands by industry have increased the need for determining the exact deviation of a machined part from being perfectly round. This need is being met by roundness measuring instruments which measure the deviation of the surface of a machined part from the axis of a very precise spindle. Although instruments for measuring out-of-roundness have been highly developed there are certain conditions that must be satisfied—for example, how to align the axis of the machined part with the axis of the precision spindle and how to differentiate between out-of-roundness and surface roughness or waviness.

RECENT demands for improved performance of machinery have brought about a need for accurately controlling the roundness of machined parts. Depending upon the eventual end use of an assembly or upon a particular component, many parts must be made round within strict tolerance limits for a number of reasons.

For example, if journal bearings are to carry high loads and to resist wear, the journal must be round. Some precision spindles which use journal bearings must have a round shaft to assure a fixed axis of rotation. Hydraulic valve spools and bodies, Diesel engine injector plungers and bushings, and hydraulic lash adjustor parts must be round to reduce the leakage area to a minimum and yet allow enough clearance so the parts can be assembled to operate satisfactorily. Ball and roller bearings must have round parts to reduce noise and vibration, to increase life, and to provide a constant torque to rotate the bearings and ensure good axial fixation. Shafts and bores subject to press fits with critical mating parts must be round to avoid distortion of the parts and, in turn, to prevent consequent degradation of performance. Surfaces that run in contact with seals must be round to avoid leakage and short seal life.

Out-of-Roundness Defined

To understand the concept of roundness as it applies to machining operations, and in particular to centerless grinding operations, an elaboration of some fundamental definitions of surfaces and surfaces of revolution is necessary together with a definition as to what is meant by out-of-roundness (Table I).

Fundamentally, the majority of machined parts having curved surfaces can have part or all of their surfaces classified as surfaces of revolution and, if perfectly made, are round. A listing of such parts and surfaces would, of course, be extremely long. A representative listing, however, can be made (Table II).

Examples of machined parts that when perfectly made are round, but are not surfaces of revolution, include: bent rollers, round wire springs (in the cross section of the wire), and curved pipe. Machined parts that are not round in terms of the definition established for out-of-roundness (Table I) include the family of parts that are helixes with representative parts such as bolt threads, coil springs, and wood screws.

Surfaces represented by oblique cylinders or oblique cones which can have round sections are not round since the axis is not perpendicular to the round section.

Out-of-Round Objects Can Have a Constant Diameter

There are many out-of-round objects that have a constant diameter in either two or three dimensions (Figs. 1, 2, and 3). These objects may have either regularly or irregularly space protuberances which may be uniform or random in size.

Many machined parts have the characteristic of being constant diameter but out-of-round. The center of the part is not conveniently available from which to measure the radius. It is for this reason, perhaps, and also because it is a relatively simple matter to measure the diameter, that many people make the false assumption that if the diameter is constant the part is round. This is an easy assumption to make because, in practice, the out-of-round effects of machined parts are not very pronounced and constant diameter parts fall in a region nearer a circle.

Out-of-roundness frequently occurs on parts that are centerless ground. Centerless grinders operate primarily on the basis of making constant diameter objects and, secondarily, on making the objects round. The less refined conditions of typical centerless grinding operations produce objects having three, five, or seven equally spaced lobes superimposed on the round surface. As the operation becomes more refined and the tolerance-versus-diameter is reduced, the surface becomes more irregular. The surface, however, is still essentially constant diameter.

Measuring Out-of-Roundness With V-Blocks and Ring Gages

It is impossible to measure the out-of-roundness of objects by measuring multiple diameters. As a result, various devices—notably the V-block and ring gage—have been devised in an attempt to determine the out-of-roundness of machined parts[1]. These devices usually consist of indicating the diameter of the part at several points when using as a reference some average of the general area being measured.

V-Blocks

To detect and measure the uniform multiple lobing effect produced on a part by some centerless grinding operations, the part is first placed in a V-block hav-

FUNDAMENTAL DEFINITIONS OF GEOMETRICAL SHAPES AND OUT-OF-ROUNDNESS

CIRCLE
A closed plane curve all points of which are equidistant from a fixed point, the center, in the plane

ROUND SURFACE
A surface such that each point on the surface lies on some cross section that is a circle. The locus of the centers of these circles is everywhere perpendicular to the plane of the cross section

SURFACE OF REVOLUTION
A round surface for which the axis is a straight line

AXIS OF A ROUND SURFACE
The locus of centers of the cross section. The axis is a straight line for surfaces of revolution

SPHERE
A surface of revolution generated by revolving a circle, the generator, about one of its diameters as the axis

RIGHT CIRCULAR CYLINDER
A surface of revolution generated about the axis by revolving a parallel straight line, the generator

RIGHT CIRCULAR CONE
A surface of revolution generated about the axis by a straight line, the generator, that intersects the axis

OUT-OF-ROUNDNESS
The out-of-roundness of the two-dimensional cross section of a three-dimensional surface may be defined as follows: Using each point within the closed two-dimensional curve as the center, it is possible to construct a largest inscribed circle and a smallest inscribed circle. There is a point for which the difference between the two radii is a minimum. That point is called the center of the cross section and the difference of the radii is called the out-of-roundness of the cross section.

Table I—Listed here are fundamental definitions of geometrical shapes and surfaces which aid in understanding the concept of roundness and the out-of-roundness of a cross section.

ing an appropriate included angle between the surfaces of the V for the number of lobes in the circumference. The part is then rotated and the region opposite the V-block is indicated with a suitable indicator.

The optimum included angle has been determined for the V-block for each of the more common lobing conditions. This angle is a 60° included angle for 3-lobed parts, a 108° included angle for 5-lobed parts, and a 128-4/7° angle for 7-lobed parts.

The V-block measurement device is most effective when (a) the parts are basically right circular cylinders, (b) the lobing is uniform, and (c) the number of lobes is known.

Ring Gages

A technique that works quite well and is useful for determining the out-of-roundness of basic right circular cylinders that are machined integrally with bulky pieces, such as the journal of a crankshaft, is to fit a split ring gage over the cylinder. If the ring gage is round and if the fit is correct, the ring gage can be rotated with respect to the cylinder being measured and the axis of the ring gage will be fixed. An indicator is then mounted to detect radial deviation of the circumference of the cylinder with respect to the ring gage.

The disadvantage of the ring gage measurement device is that each size piece to be measured requires an associated ring gage that must be accurately made.

Highest Accuracy Obtained From Precision Spindle Instruments

During the past decade, new instruments have been developed specifically for measuring out-of-roundness. These instruments, referred to as precision spindle instruments, have proven to be most effective for accurately measuring any part having a round surface.

The essential components of a precision spindle instrument consist of a very precise spindle to provide a reference center, a centering device to align the axis of the spindle with the axis of the part being measured, and a suitable indicator which may be coupled with a recorder to measure the deviations of the part from round.

There are two basic types of precision spindle instruments for measuring out-of-roundness. The first type has the part to be measured rotated by the precision spindle (Fig. 4-left). In the second type of instrument, the indicator is rotated about the part being measured (Fig. 4-right). Each type has its advantages and disadvantages. The choice between the two must, in part, take into account the type of piece being measured.

A roundness measuring instrument of the precision spindle type has been designed and built by the GM Research Laboratories. Called the GM Roundicator (Fig. 5), it is of the type shown in Fig. 4-right and can accommodate machined parts ranging in size from small needle bearing rollers to cylinder bores of Diesel engines. The axis of the lapped spindle is held concentrically and axially to less than two millionths of an inch by specially designed frictionless hydrostatic bearings.

Circular Chart Shows Shape of Out-of-Roundness Detail

Since it is important to know the shape of the out-of-roundness detail, the output of precision spindle instruments usually is in the form of a graph. The graph may be recorded on either a circular chart (Appendix) or a strip chart.

MACHINED PARTS HAVING SURFACES OF REVOLUTION

SPHERICAL SURFACES
Ball Bearing Balls
Spherical Rocker Arm Seats
Spherical Check Valve Seat
Ball Joint Suspension Balls

RIGHT CIRCULAR CYLINDER SURFACES
Crankshaft Journals
Pipe
Bored Holes
Hydraulic Valve Spools
Piston Pins
Straight Roller Bearing Rollers
Shafts

RIGHT CIRCULAR CONE SURFACES
Tapered Roller Bearing Rollers
Tapered Roller Bearing Races
Poppet Valves
Reamed Holes
Glass Stop Cocks

OTHER SURFACES OF REVOLUTION
Ball Bearing Races
Spherical Roller Bearing Rollers
Labyrinth Seals
'O' Rings

Table II—This is a representative summary of machined parts having surfaces which can be classified as surfaces of revolution.

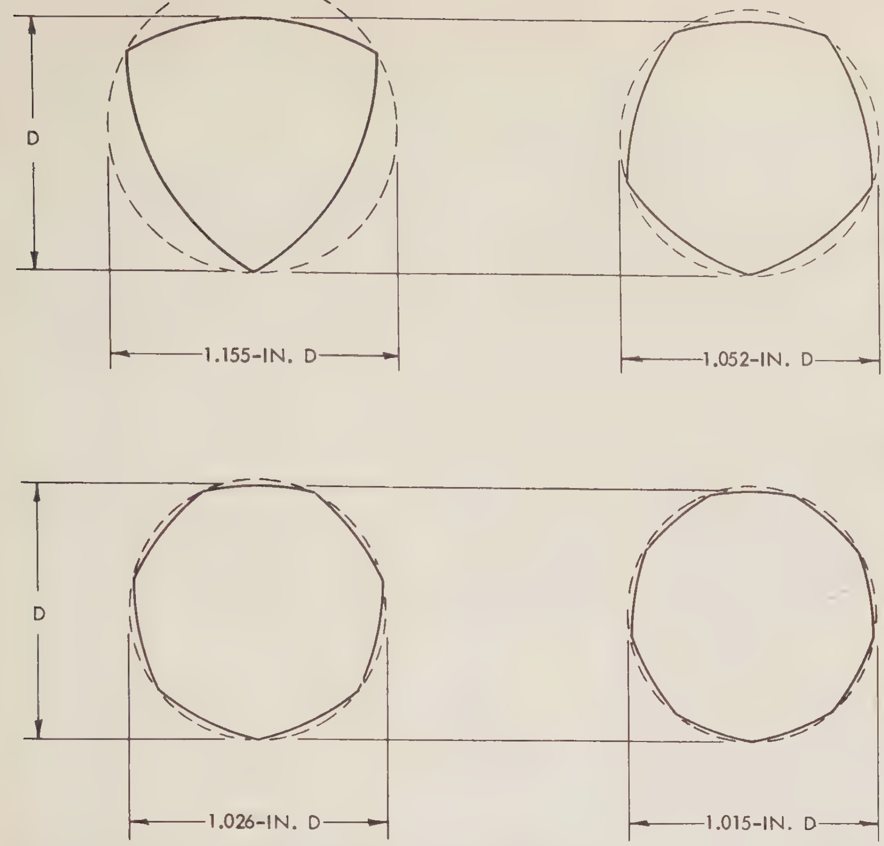

Fig. 1—Shown here are extreme cases of iso-diametric, multiple sided objects that are of constant diameter *D* in two dimensions, as might be measured by a vernier caliper or height gage, but are not round by the definition given in Table I. For these two-dimensional objects, the number of undulations is odd. Each corner is the center of the radius for the opposite side and the radius is equal to the diameter of the object. In these extreme cases for objects of one-in. diameter, the smallest circle that can encompass the object is 1.155 in. diameter for three undulations, 1.052 in. diameter for five undulations, 1.026 in. diameter for seven undulations, and 1.015 in. diameter for nine undulations.

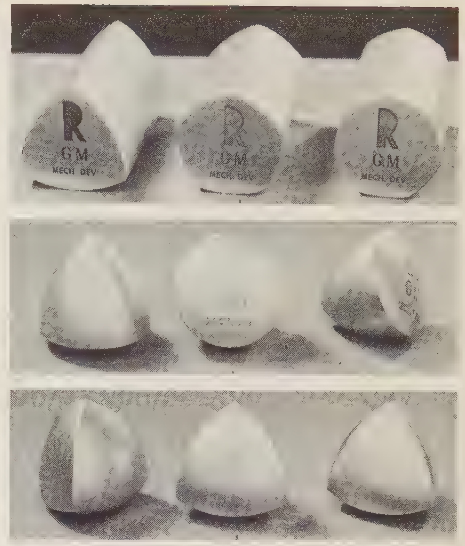

Fig. 3—An example of iso-diametric, out-of-round objects is shown at the top. These constant diameter "rollers," when placed between a heavy book and a flat surface, would give the book a deceptively smooth motion, since the book would move in a horizontal plane with no vertical component of motion.

As might be expected, there also are three-dimensional, out-of-round objects of constant diameter in three dimensions (center). These objects, when observed on their axis, look like a circle. When the objects are observed perpendicular to their axis, they have the characteristic shape of the object shown at the top-left in Fig. 1.

It might be expected, also, that a tetrahedron with curved faces would be a constant diameter object. The diameter, however, is not quite constant. Three of the six edges must be modified with a radius to give a constant diameter. The iso-diametric tetrahedrons shown at the bottom have the modified edges. Either of these shapes, when used as balls between two flat surfaces, allow very smooth two-dimensional motion of one surface with respect to the other. The objects do not make good ball bearing balls, however, since the center of gravity is constantly changing and some areas are more highly loaded than when using a ball.

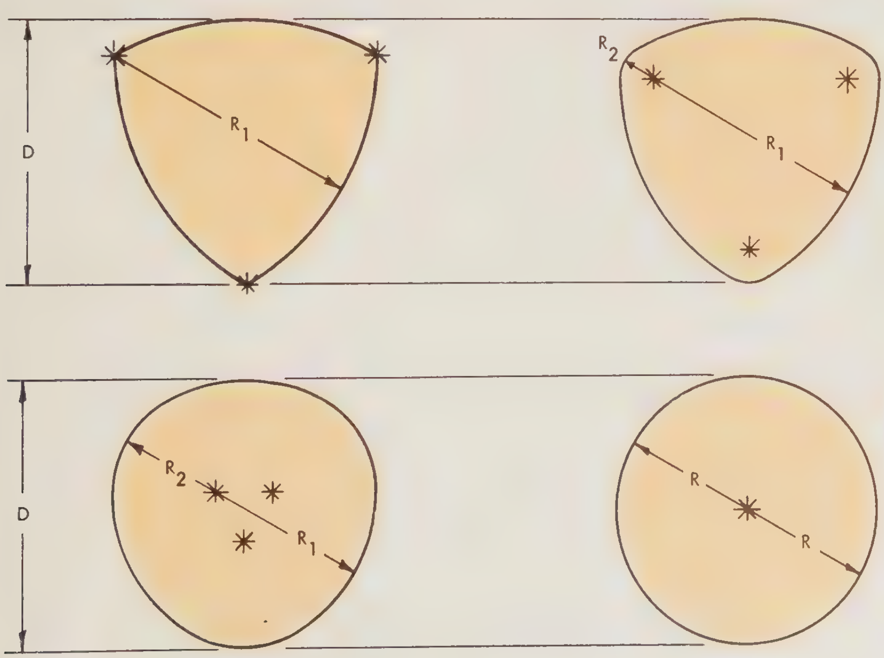

Fig. 2—In practice, the out-of-round effects are not so pronounced and the constant diameter parts fall in a region nearer a circle, as illustrated by these objects which show the transition of out-of-round to round iso-diametric objects. If $R_1 = 0$, the most severe case is present. If $R_1 = R_2$, the object is a circle. The diameter of the object is equal to $R_1 + R_2$.

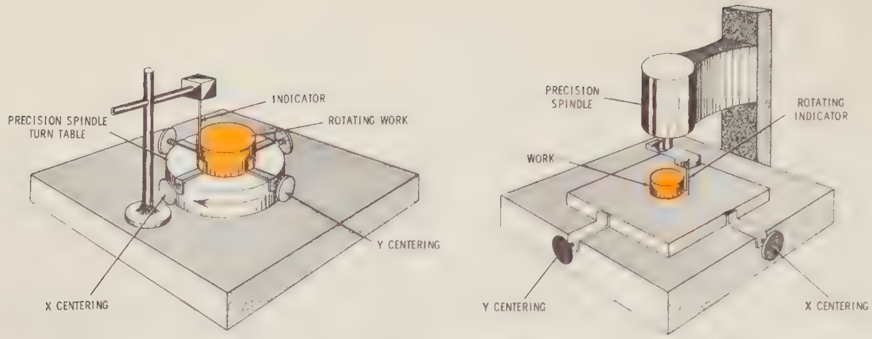

Fig. 4—Precision spindle instruments for measuring out-of-roundness deviations are of two basic types. The instrument shown at the left is of the type in which the part being measured is rotated by the precision spindle. Since the axis of the part must coincide with the rotational axis of the spindle, the centering mechanism must be placed between the part and the spindle so that the centering device rotates with the spindle. The indicator is adjusted to determine conveniently how nearly the part is a surface of revolution as well as to measure out-of-roundness. This type of precision spindle instrument is limited to measuring parts that weigh less than the spindle can support.

The type of precision spindle instrument shown at the right rotates the indicator about the work. Provision is made to adjust the indicator to accommodate the size of the part being measured. The centering mechanism is stationary. The size of the part being measured can be large and heavy and is limited only by its physical size with respect to the clearances of the machine. Measurements of the deviation from a surface of revolution can be made, but with more difficulty than out-of-roundness.

The chart produced by precision spindle instruments is highly distorted. The height of the out-of-round irregularities is much more highly magnified than their width. It also is necessary for the zero to be depressed to the extent that the deviations from round occupy a considerable amount of the chart width. This is done to make the deviations easily detectable and to allow the chart paper to be of reasonable size. Since the chart is quite distorted, certain fundamental concepts must be applied to determine the shape of the part when evaluating its chart.

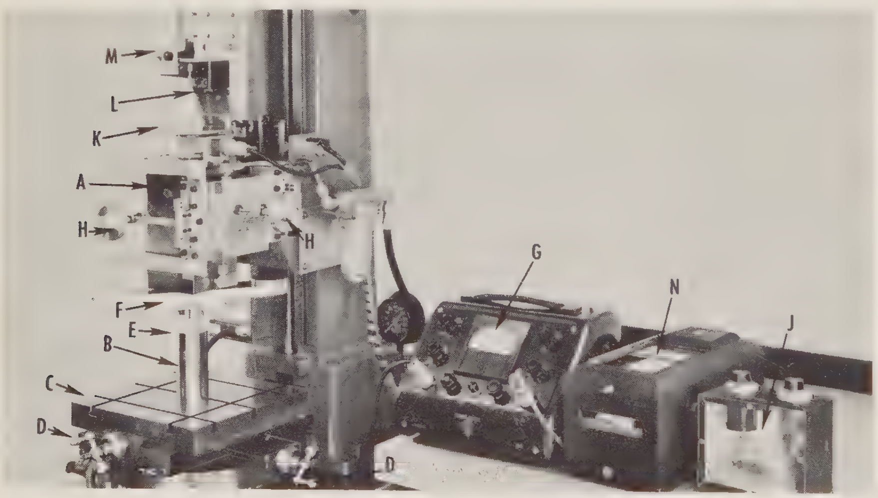

Fig. 5—The GM Roundicator shown here is a precision spindle instrument for measuring roundness. Designed and built by the GM Research Laboratories, the Roundicator detects roundness deviations of less than two millionths of an inch on parts up to 18 inches in diameter. A lapped spindle is held concentrically and axially by hydrostatic bearings located in a housing *A*.

The procedure used to measure the roundness of a part is as follows. The part to be measured *B* is placed on an adjustable table *C*. The axis of the part is adjusted to be approximately coaxial with the spindle by means of coarse adjusting controls *D*. A stylus *E* is then engaged with the part by appropriately adjusting a stylus carriage *F*. Radial motion of the stylus is indicated on the meter *G*. Further manipulation of controls *D* and *F* is made to align the part with the spindle to an accuracy of about 0.001 in. Fine centering to millionths of an inch is done by turning micrometer adjusting screws *H* which displace the housing *A* through a 25 to 1 lever system using frictionless spring pivots. Fine compensating adjustments to center the needle on meter *G* for the stylus radial position are made by box *J*. After the adjustments have been made for minimum meter variation, a circular paper is placed on a platen *K*. A flexible drive shaft *L* is then lowered into position and a drive motor switch *M* is turned on. A recording pen is engaged for one revolution of the spindle assembly. The resulting chart is a Roundigram, which is a circular chart recording of the radial variations of the part from being perfectly round. If closer inspection of surface finish is desired, a strip chart having greater circumferential magnification may be made on recorder *N*.

Circular Chart

The circular chart, when used for recording out-of-roundness, has certain properties that have been elaborated upon by R. E. Reason[2]. These are as follows:

- The angular relationship of the protuberances of the part is preserved with respect to the center of the chart
- In the event that the part is not perfectly centered in the instrument, either best fitting circles can be constructed on the chart according to the out-of-roundness definition or a circular template may be fitted with very little error introduced for measurements taken in a radial direction. The maximum radial error is on the order of one-half per cent of the radius of the recording, if the eccentricity of the fitted circle with respect to the chart is 10 per cent of the recording, and two per cent error with 20 per cent eccentricity
- Differences in diameter of the part are determined by measuring the differences in the diameter of the recording through the center of the chart. This is true regardless of how poorly the part is centered in the instrument.

Strip Chart

The strip chart has the property that its horizontal amplification is not inherently limited, as is the case with the circular chart. This property of the strip chart is useful when it becomes necessary to examine fine roughness detail on the surface of the part.

Conclusions

The definition given for out-of-roundness (Table I) has the following advantages:

- When measured from the defined center, the out-of-roundness can be determined conveniently from a circular chart, strip chart, or displacement indicator
- The definition is amenable to instrumentation which is either manually or automatically operated
- The minimum amount of metal that must be removed from the part is immediately evident.

Although instruments for measuring out-of-roundness have been developed to the point where roundness deviations on the order of millionths of an inch can be measured, certain areas relating to out-

of-roundness measurement work still remain to be investigated such as:

- How to differentiate between out-of-roundness, waviness, and surface roughness
- How to define the out-of-roundness of a part which is basically the segment of a circle
- How to specify the type and frequency of lobing that is particularly undesirable in round parts which must perform under critical operating conditions.

There is one final point which should be considered in regard to out-of-roundness. The engineering drawing serves as the means for communicating to those in manufacturing how a part should be made. The relatively recent adoption of standardized numerical symbolism and terminology for specifying surface finish, for example, eliminated confusion as to the limits within which a machinist must work. The question can be asked, therefore: Should a standard symbol be adopted for industry-wide use to designate on engineering drawings the allowable out-of-roundness of the part?

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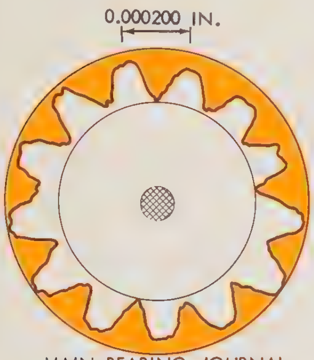
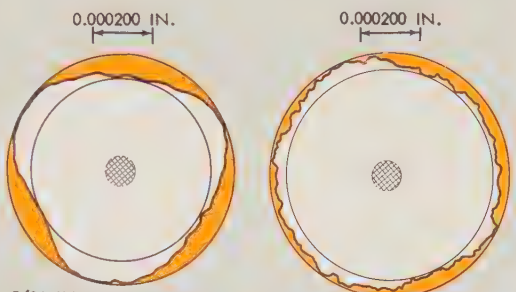
Appendix

The circular charts shown in this Appendix are Roundigrams produced by the GM Roundicator—a precision spindle instrument used to measure the out-of-roundness of machined parts.



MAIN BEARING JOURNAL

Roundigram shows two distinct types of protuber- that contribute to out-of-roundness—the two per ution, which some investigators refer to as ovality, he 13 per revolution, which some might refer to as ness. The type having 13 protuberances per revolu- is impossible to detect by measuring the diameter, s far more serious in reducing the load carrying city of a journal bearing for equal out-of-roundness ibution than the two protuberance pattern.

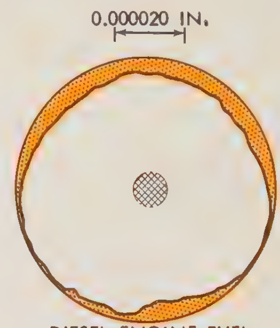


5/16-IN. DIAM DOWEL PIN

PISTON PIN

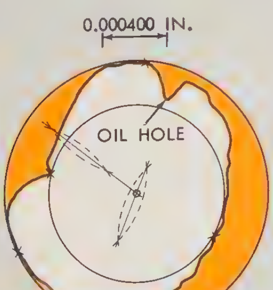
These two Roundigrams show traces produced from parts which have been centerless ground. The chart at the left is an example of the work produced by a centerless grinder that has not been adjusted to remove the uniform lobing characteristics of many machined parts. The out-of-roundness is quite small in terms of conventional shop practice, being on the order of 0.000080 in. In the Roundigram at the right, the characteristic of the surface is quite random, which is typical of parts produced by a centerless grinder that has been adjusted to remove the uniform lobing property.

This Roundigram (below) shows how a chart might be used to study the leakage area of an out-of-round piston fitted into a round hole. The dotted area represents the leakage path. The out-of-roundness of this particular part is only six microinches.



DIESEL ENGINE FUEL INJECTOR PLUNGER

Reproduced on this Roundigram is the construction required to locate the center of an irregularly shaped object. The out-of-roundness is the difference between the radii of the inscribed and the circumscribed circles.

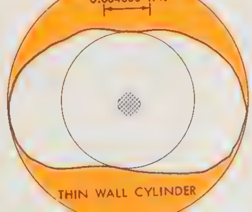


CONNECTING ROD JOURNAL

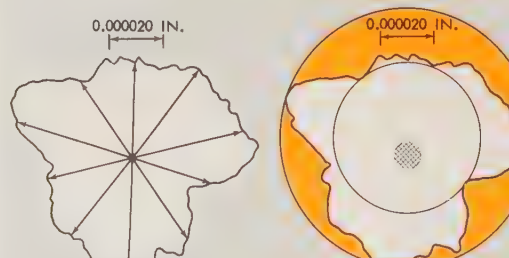
MAIN BEARING JOURNAL

A record of a journal bearing that included an oil pressure supply hole is illustrated by this Roundigram. Since the out-of-roundness of only the journal surface is of interest, the effects of the oil hole are ignored.

Of all the Roundigrams shown in this Appendix, the part shown below is the only one where the out-of-roundness could be measured by making multiple measurements of the diameter. Ovality is quite typical of thin walled cylinders.



THIN WALL CYLINDER



1 IN. MICROMETER STANDARD

1 IN. MICROMETER STANDARD

The Roundigram at the left is an example of a trace made at a very high magnification from indicator to recorder. The differences in diameter can be determined by measuring the difference in the length of arrows which pass through the center of the recording. The diameter of this part, which is a micrometer testing gage, is constant to a total spread of 6 parts in 10^6. The Roundigram at the right shows that the out-of-roundness is considerably higher, being on the order of 40 parts in 10^6 when referred to the radius of the gage. Note that this particular trace is not centered with respect to the chart. The error introduced in the radial measurements by not centering perfectly is quite small and, in this particular example, is less than one microinch.

Analyzing the Exhaust Velocity Requirements for Electrically Powered Rockets

Included in the basic studies made by Allison Division research engineers on advanced propulsion systems for space craft was a study made on the exhaust velocity requirements of electrically powered rockets. Of specific interest was the relationship between the exhaust velocity and the ratio of payload to take off mass. The purpose of this study was to determine the exhaust velocity which would result in the maximum payload mass fraction for possible combinations of vehicle characteristics and mission requirements. An equation giving optimum exhaust velocity (specific impulse) of an electrically powered rocket was derived by maximizing the ratio of payload mass to take off mass. This equation was then used in a study of three basic rockets. The rockets were analyzed over a range of mission velocities and thrust times which approximated various space maneuvers and trips to the near planets of Mars and Venus. The results of the study, which was made in early 1958 and which was typical of the work done by Allison in the early stages of its space research program, showed that maximum payload to take off mass ratios for electrically powered rockets could be obtained at exhaust velocities substantially lower than those generally considered for electric rockets at that time.

In recent years, the electrically powered space rocket has received increased attention as a possible future vehicle for extended types of space missions, such as manned flights to the near planets of Mars and Venus. As part of a broad energy conversion research program, Allison Division has been engaged in various studies on this advanced type of rocket propulsion system. Allison research engineers, like others, have explored the many requirements of the electrically powered rocket during recent years to gain new information and to contribute to the state of the art.

One example of the type of study that has been conducted at Allison to gain new information was an analysis made on rocket exhaust velocity (specific impulse*) and its relationship to the ratio of payload mass to take off mass. At the time the analysis was made in early 1958, prevailing opinion held that increases in exhaust velocity to values in the range of 100,000 to 200,000 m per sec, or even higher, would result in increased payload to initial mass ratio for electrical rockets. The results of the analysis, to be discussed, showed that maximum payload mass to take off mass ratios could be obtained at lower exhaust velocities than were generally considered for electric rockets at that time. The analysis, therefore, was of considerable value to Allison research engineers in specifying the future requirements of electrical propulsion devices.

*The terms *exhaust velocity* and *specific impulse* are used interchangeably in this paper. See *Appendix A* to this paper for a mathematical derivation showing the relationship between rocket exhaust velocity and specific impulse.

Electrically Powered Rocket Could Carry Heavier Payload

The electrically powered rocket represents one way to obtain high specific impulse with a low mass ratio—that is, the ratio of the take off mass to the final mass of the rocket. This gives the electrically powered rocket an advantage over the present chemically powered rocket in that a greater amount of payload can be carried in relation to the over-all weight of the rocket.

Various methods have been proposed for obtaining electrically powered rockets. Stuhlinger[1,2] and others have proposed to accelerate metallic ions to produce microthrust in an orbiting space rocket. Such a rocket, frequently referred to as an ion rocket, would gradually escape from the gravitational field of the Earth and would be able to travel as far as the near planets. Plasma propulsion units such as magnetohydrodynamic accelerators[3] and arc-jets[4] also have been proposed as possible high energy electrical rocket motors.

In most of the studies, it has been assumed that a nuclear reactor-generator will provide the power to accelerate the rocket propellant and that a heat rejection system, or radiator, will be used to reject the excess heat resulting from energy conversion inefficiency. The mass of the nuclear reactor-generator system will be, to a good approximation, proportional to its steady state operating power level. The mass of the radiator system will be a direct function of the total efficiency of the energy converters and the power level at which they operate.

The power demanded by the propulsion system will be proportional to the square of the exhaust velocity and will be directly related to the propellant flow rate. It is evident, then, that the mass of the nuclear reactor-generator system and the radiator system will be proportional in the same manner to the exhaust velocity and propellant flow rate, since the propulsion system of the electrical rocket will be the major consumer of power.

Equation for Optimum Exhaust Velocity Derived

It is desirable in space craft that the payload mass be as large as possible in relation to its total mass. For a given space mission, therefore, there is an optimum operating condition of the electrical rocket engine which will minimize the mass of the nuclear reactor-generator and the radiator and other masses which do not contribute to the useful payload.

The approach used by Allison research engineers to arrive at an optimum operating condition was to derive an equation for optimum exhaust velocity by maximizing the ratio of payload mass to take off mass.

It had been shown[5] that if the exhaust velocity of a rocket was held constant

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Allison Division

How high must exhaust velocity be to obtain maximum payload mass?

throughout the acceleration period, the velocity change of the rocket in free space would equal

$$\Delta V = (c)\, ln \left(\frac{M_1}{M_2}\right) \qquad (1)$$

where

ΔV = total change in rocket velocity, that is, mission velocity (m per sec or ft per sec)

c = exhaust velocity of the rocket (m per sec or ft per sec)

$\frac{M_1}{M_2}$ = mass ratio of the rocket

M_1 = take off mass of the rocket (kg or slugs)

M_2 = final mass of the rocket (kg or slugs).

The mass ratio of the rocket also could be written as

$$\frac{M_1}{M_2} = 1 + \frac{M_P}{M_2} \qquad (2)$$

where

M_P = mass of the rocket propellant (kg or slugs).

If the final mass of the rocket included only the useful payload mass M_{PL}, the exhaust velocity c should be as large as possible to minimize the mass of the propellant. The final mass of the rocket, however, would also include the following masses:

- The mass of the powerplant, M_{PP} (kg or slugs)
- The mass of the heat rejection system, M_R (kg or slugs)
- The mass of the structure needed to support the payload, M_S (kg or slugs).

The masses M_{PP} and M_R are nearly proportional to the maximum power level at which they operate. Also, the mass M_S is a direct function of the payload mass.

The ratio of the payload mass to the take off mass for a given electrical rocket and space mission could be expressed by the following equation (Appendix B):

$$\frac{M_{PL}}{M_1} = \frac{1}{1 + K_1}\left[e^b - \frac{c^2}{2t}(1 - e^b)\left\{\frac{1}{\eta}(K_2 + K_3) - K_2\right\}\right] \qquad (3)$$

where

M_{PL} = payload mass of the rocket (kg or slugs)

K_1 = $\frac{M_S}{M_{PL}}$ (kg per kg or slugs per slug)

K_2 = $\frac{M_R}{P_T - P_J}$ (kg per watt or slugs per hp)

K_3 = $\frac{M_{PP}}{P_T}$ (kg per watt or slugs per hp)

$P_T - P_J$ = rejected power

P_T = total thermal power available (watts or hp)

P_J = jet exhaust power (watts or hp)

b = $-\frac{\Delta V}{c}$

t = thrust time (sec)

η = $\frac{P_J}{P_T}$ (watt per watt or hp per hp).

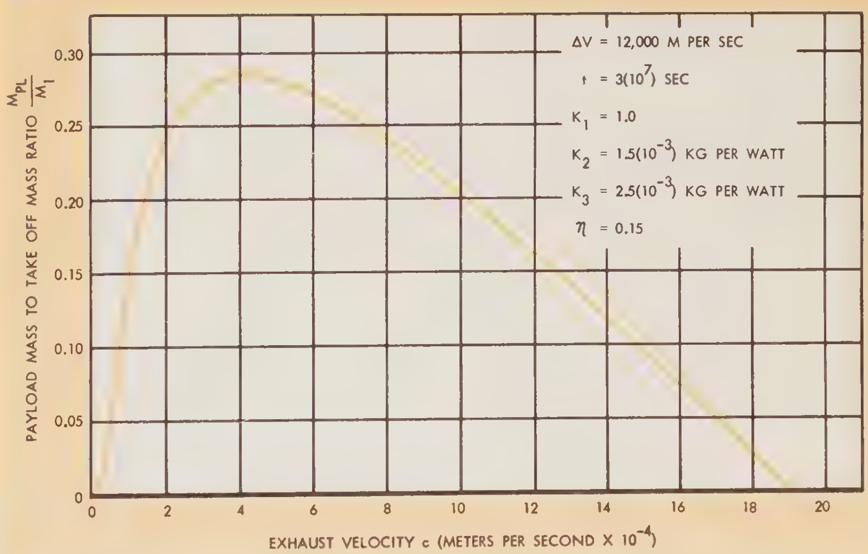

Fig. 1—This is a plot based on equation (3) for fixed values of mission velocity ΔV, thrust time t, and efficiency ratios K_1, K_2, K_3, and η. The plot was made to determine the maximum payload to take off mass ratio and exhaust velocity required for a typical space mission. For the parameters chosen, a maximum ratio of 0.284 occurred at an exhaust velocity of 43,000 m per sec.

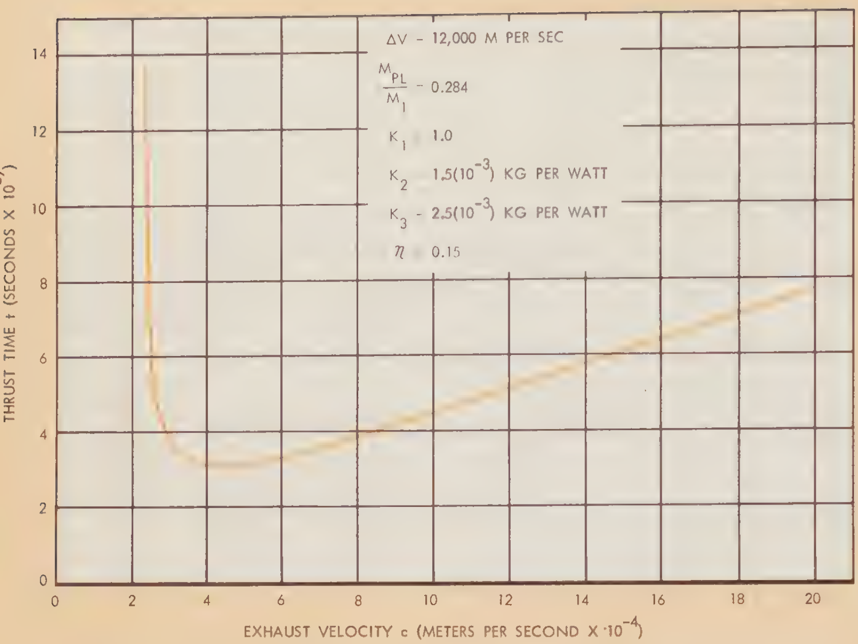

Fig. 2—This plot was made to determine if a shorter thrust time t could be achieved with the same payload to take off mass ratio found in Fig. 1. The plot showed that an exhaust velocity of 43,000 m per sec again gave optimum operation for the same thrust time used when plotting Fig. 1. Any other exhaust velocity would give a longer thrust time.

The development of equation (3) made it possible to investigate the relationship between exhaust velocity and the ratio of payload mass to take off mass. Typical fixed values for mission velocity, thrust time, and efficiency ratios were substituted into equation (3) and a plot made of exhaust velocity versus payload mass to take off mass ratio (Fig. 1). A maximum payload mass to take off mass ratio of 0.284 occurred at an exhaust velocity of 43,000 m per sec. To determine if this same payload mass to take off mass ratio could be achieved with a shorter thrust time than that assumed for the plot in Fig. 1, another plot was made of thrust time versus exhaust velocity for the same mission velocity, efficiency ratios, and a payload mass to take off mass ratio of 0.284 (Fig. 2). It was again demonstrated that an exhaust velocity of 43,000 m per sec gave optimum operation. Any other exhaust velocity resulted in a longer thrust time. On such a space mission, a 100-ton rocket of this design could have a payload of 28.4 tons. The remaining mass of 71.6 tons would be comprised of structure, propellant, heat rejection system, and power plant.

The general optimum exhaust velocity was found by partially differentiating equation (3) with respect to c and setting the equation equal to zero (Appendix C). This gave the following equation:

$$\frac{\Delta V t}{c_o^3\left[\frac{1}{\eta}(K_2 + K_3) - K_2\right]} + \frac{\Delta V}{2c_o} + 1 = e^d \qquad (4)$$

where

c_o = optimum exhaust velocity (m per sec or ft per sec)

$$d = \frac{\Delta V}{c_o}.$$

The mathematical validity of equation (4) is discussed in Appendix C, where it also is shown that for c_o to correspond to a maximum payload mass to take off mass ratio (as opposed to a minimum ratio), the following equation also must be satisfied:

$$\frac{c_o^2}{2t}\left(1 + \frac{\Delta V}{c_o}\right)\left[\frac{1}{\eta}(K_2 + K_3) - K_2\right] + \frac{\Delta V}{c_o} < 3. \qquad (5)$$

Three Basic Space Rockets Studied

With the establishment of equations (3) and (4) a study then was made on three basic electrically powered space rockets. A digital computer was used as an aid in solving equations (3) and (4).

The three rockets studied were assumed to be powered by turbo-electric generators driven by the heat energy from a conventional nuclear reactor. The excess heat resulting from turbine losses, generator inefficiency, and electrical losses was assumed to be rejected from the rocket by a radiator system.

The three electrically powered space rockets studied were referred to as the *optimistic rocket*, the *compromise rocket*, and the *pessimistic rocket*. The following is a summary of efficiency ratios K_1, K_2, K_3, and η assumed for each rocket.

Optimistic Rocket

K_1 = 0.5 kg per kg

K_2 = 0.75(10^{-3}) kg per watt

K_3 = 1.25(10^{-3}) kg per watt

η = 0.30 watt per watt.

The optimistic rocket postulated that considerable progress would be made in the development of presently feasible nuclear reactor-generators, radiators, and structural materials.

Compromise Rocket

K_1 = 1.0 kg per kg

K_2 = 1.5(10^{-3}) kg per watt

K_3 = 2.5(10^{-3}) kg per watt

η = 0.15 watt per watt.

The compromise rocket was based on efficiency ratios for K_1, K_2, K_3, and η which appeared possible at the time of the analysis.

Pessimistic Rocket

K_1 = 2.0 kg per kg

K_2 = 3.0(10^{-3}) kg per watt

K_3 = 5.0(10^{-3}) kg per watt

η = 0.075 watt per watt.

The pessimistic rocket represented a system in which projected efficiency ratios at that time for K_1, K_2, K_3, and η were not attained.

The three space rockets were analyzed over a range of thrust times for mission velocities of 3,500 m per sec, 12,000 m per sec, and 24,000 m per sec (Appendix D). These mission velocities were selected to correspond approximately to a simple escape maneuver from an Earth orbit, a one-way flight to an orbit of Mars or Venus, and a round trip between Earth and Mars or Venus.

The three electrically powered space rockets were further analyzed in a study based on the rockets making a round trip from Earth to Mars. A 1,000-nautical mile circular orbit of the Earth to a 1,000-nautical mile orbit of Mars was assumed with a thrust time of two years (Table I).

Conclusion

At the present time, the electrically propelled rocket appears to offer a feasible answer to a space vehicle capable of undertaking missions requiring heavy payloads for extended expeditions to the near planets. Much more knowledge and experience must be obtained, however, before such a space vehicle becomes a reality. This knowledge and experience is being gained in many research programs now in existence aimed at developing more information on the many details pertaining to its operation.

The analysis discussed here is but one example of the work which has been done by Allison research engineers in the search for new information pertinent to solving the many inherent technical problems of space flight. In this particular case, information was developed which showed that maximum payload to take off mass ratios for electrically propelled rockets could be obtained at substantially lower exhaust velocities than had been reported in the technical literature at the time the analysis was made. The analysis indicated that a specific impulse of about 6,000 sec would be optimum for a nuclear powered rocket undertaking a two-year round trip to Mars (Table I).

Allison Division is actively engaged in various studies pertaining to space technology. Such studies include space mission analysis, ionic and plasma propulsion devices, nuclear power supplies, direct conversion devices, closed cycle mechanical power units, solar heat collectors and heat storage units, heat rejection systems, high energy chemical propellants, and rocket engine cases and nozzles. The contributions made by Allison research engineers in these fields are typical of the work being done by others toward the development of propulsion systems which ultimately may provide the means for extended space travel.

	ROUND TRIP TO MARS		
	OPTIMISTIC ROCKET	COMPROMISE ROCKET	PESSIMISTIC ROCKET
Optimum Exhaust Velocity	133,000 m per sec	59,000 m per sec	21,000 m per sec
Specific Impulse	13,600 sec	6,000 sec	2,140 sec
Ratio of Payload Mass to Take Off Mass	0.463	0.215	0.025
Ratio of Propellant Mass to Take Off Mass	0.165	0.335	0.681
Mission Velocity	24,000 m per sec	24,000 m per sec	24,000 m per sec
Safety Factor	25 per cent	25 per cent	25 per cent

Table I—This table summarizes the results of an analysis made on the three rockets for a round trip from a 1,000-nautical mile circular orbit of the Earth to a 1,000-nautical mile orbit of Mars. A thrust time of two years was assumed. The term *safety factor* refers to the mission velocity over that of a minimum energy transfer between planets. A value of 25 per cent was chosen to compensate for losses from trajectory corrections and so-called gravitational losses in flight[6].

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Appendix A

Relationship Between Rocket Exhaust Velocity and Specific Impulse

Specific impulse is defined by the following relationship:

$$I_{sp} = \frac{F}{w} \quad (1A)$$

where

I_{sp} = specific impulse of jet (sec)

F = rocket thrust (newtons or lb)

w = weight rate of propellant flow (newtons per sec or lb per sec).

Rocket thrust F is defined as

$$F = mc \quad (2A)$$

where

m = mass rate of propellant flow (kg per sec or slugs per sec)

c = rocket exhaust velocity (m per sec or ft per sec).

The mass rate m of propellant flow, in turn, is defined as

$$m = \frac{w}{g_c} \quad (3A)$$

where

g_c = acceleration of gravity at surface of the Earth (m per sec^2 or ft per sec^2).

Specific impulse, therefore, may be expressed by the following relationship:

$$I_{sp} = \frac{c}{g_c} \quad (4A)$$

where

g_c = 9.8 m per sec^2 = 32.2 ft per sec^2.

Appendix B

Derivation of Payload to Take Off Mass Ratio

The total change in rocket velocity (mission velocity) ΔV, in m per sec or ft per sec, is defined as

$$\Delta V = (c) ln \left(\frac{M_1}{M_2}\right) \quad (1B)$$

where

c = exhaust velocity of the rocket (m per sec or ft per sec)

M_1 = take off mass of the rocket (kg or slugs)

M_2 = final mass of the rocket (kg or slugs).

The take off mass M_1 of the rocket is comprised of

$$M_1 = M_P + M_2 \quad (2B)$$

where

M_P = mass of the rocket propellant (kg or slugs).

The final mass M_2 of the rocket is comprised of

$$M_2 = M_{PL} + M_S + M_R + M_{PP} \quad (3B)$$

where

M_{PL} = mass of the payload (kg or slugs)

M_S = mass of the structure needed to support the payload (kg or slugs)

M_R = mass of the heat rejection system (kg or slugs)

M_{PP} = mass of the power plant (kg or slugs).

By definition:

$$K_1 = \frac{M_S}{M_{PL}}$$

$$K_2 = \frac{M_R}{P_T - P_J}$$

$$K_3 = \frac{M_{PP}}{P_T}$$

$$\eta = \frac{P_J}{P_T}$$

where

P_T = total thermal power available (watt or hp)

P_J = jet exhaust power (watt or hp).

Equation (3B) can be rewritten as follows:

$$M_2 = M_{PL} + K_1(M_{PL}) + K_2(P_T - P_J) + K_3(P_T)$$

or,

$$M_2 = M_{PL}(1 + K_1) + P_J\left[\frac{1}{\eta}(K_2 + K_3) - K_2\right]. \quad (4B)$$

Let

$$A = \frac{1}{\eta}(K_2 + K_3) - K_2.$$

Equation (4B), therefore, becomes:

$$M_2 = M_{PL}(1 + K_1) + (P_J)(A). \quad (5B)$$

The equation for the total change in rocket velocity (equation 1B) may be rewritten as follows:

$$\frac{M_1}{M_2} = e^a$$

or,

$$\frac{M_2}{M_1} = e^b \quad (6B)$$

where

$$a = \frac{\Delta V}{c}$$

$$b = -\frac{\Delta V}{c}.$$

Equation (5B) may be substituted into equation (6B) to give

$$\frac{M_{PL}(1 + K_1) + (P_J)(A)}{M_1} = e^b. \qquad (7B)$$

The jet exhaust power P_J may be expressed as

$$P_J = \frac{1}{2}(mc^2)$$

or

$$P_J = \frac{1}{2}\left(\frac{M_P}{t}\right)c^2 \qquad (8B)$$

where

m = mass rate of propellant flow (kg per sec or slugs per sec)

c = rocket exhaust velocity (m per sec or ft per sec)

t = thrust time (sec).

Substituting equation (8B) into equation (7B) gives

$$\frac{M_{PL}(1 + K_1)}{M_1} + \frac{(M_P)(c^2)(A)}{2(M_1)(t)} = e^b \quad (9B)$$

But, from equations (1B) and (2B):

$$1 - \frac{M_P}{M_1} = e^b. \qquad (10B)$$

Substituting equation (10B) into equation (9B) and simplifying gives

$$\frac{M_{PL}}{M_1} = \frac{1}{1 + K_1}\left[(e^b) - \left(\frac{c^2A}{2t}\right)\left(1 - e^b\right)\right]. \qquad (11B)$$

In certain calculations, the payload to propellant mass ratio may be useful. This ratio, obtained from equation (10B) and (11B) is:

$$\frac{M_{PL}}{M_P} = \frac{1}{1 + K_1}\left[\frac{1}{e^a - 1} - \frac{c^2A}{2t}\right]. \qquad (12B)$$

Appendix C

Optimization of Exhaust Velocity

The equation derived in Appendix B for the payload mass to take off mass ratio was

$$\frac{M_{PL}}{M_1} = \frac{1}{1+K_1}\left[(e^b) - \left(\frac{c^2A}{2t}\right)\left(1 - e^b\right)\right].$$

Partially differentiating this equation with respect to c gives:

$$\frac{\partial\left(\frac{M_{PL}}{M_1}\right)}{\partial c} = \frac{1}{1+K_1}\left[(e^b)\left(\frac{\Delta V}{c^2} + \frac{\Delta VA}{2t} + \frac{Ac}{t}\right) - \frac{Ac}{t}\right]. \qquad (1C)$$

To find values of c which maximize the ratio of the payload mass to the take off mass, equation (1C) is set equal to zero. Cancelling and rearranging terms gives the following equation:

$$\frac{\Delta Vt}{(c_o{}^3)(A)} + \frac{\Delta V}{2c_o} + 1 = e^d \qquad (2C)$$

where

$$d = \frac{\Delta V}{c_o}.$$

To determine if the values of c_o found from equation (2C) actually represent maximums of the payload mass to take off mass ratio (as opposed to minimum values), the second partial derivative of the payload mass to take off mass ratio equation must be taken with respect to c as follows:

$$\frac{\partial\left(\frac{M_{PL}}{M_1}\right)}{\partial c^2} = \frac{1}{1+K_1}\left[(e^b)\left(\frac{\Delta V^2}{c^4} - \frac{2\Delta V}{c^3} + \frac{\Delta V^2A}{2tc^2} + \frac{\Delta VA}{tc} + \frac{A}{t}\right) - \frac{A}{t}\right]. \qquad (3C)$$

In order that the ratio of the payload mass to the take off mass be a maximum, equation (3C) must be less than zero, or

$$e^b\left(\frac{\Delta V^2}{Ac^4} - \frac{2\Delta V}{Ac^3} + \frac{\Delta V^2}{2tc^2} + \frac{\Delta V}{tc} + \frac{1}{t}\right) < \frac{1}{t}. \qquad (4C)$$

Substituting equation (1C) into equation (4C) gives:

$$\frac{Ac_o{}^2}{2t}\left(1 + \frac{\Delta V}{c_o}\right) + \frac{\Delta V}{c_o} < 3. \qquad (5C)$$

It also can be shown mathematically that if equations (2C) and (5C) are satisfied, the optimum exhaust velocity c_o corresponds to the only maximum payload mass to take off mass ratio for given values of ΔV, A, and t.

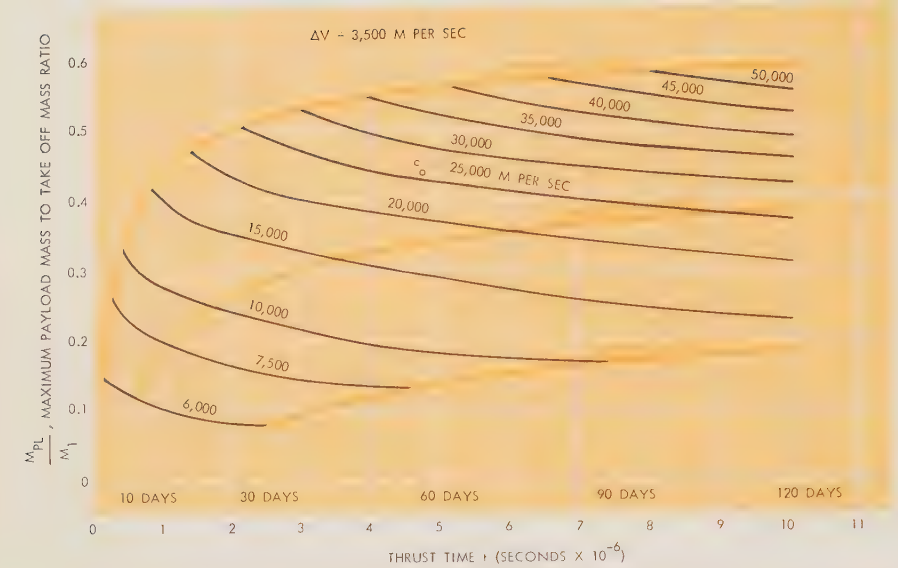
Appendix D

Optimum Exhaust Velocities

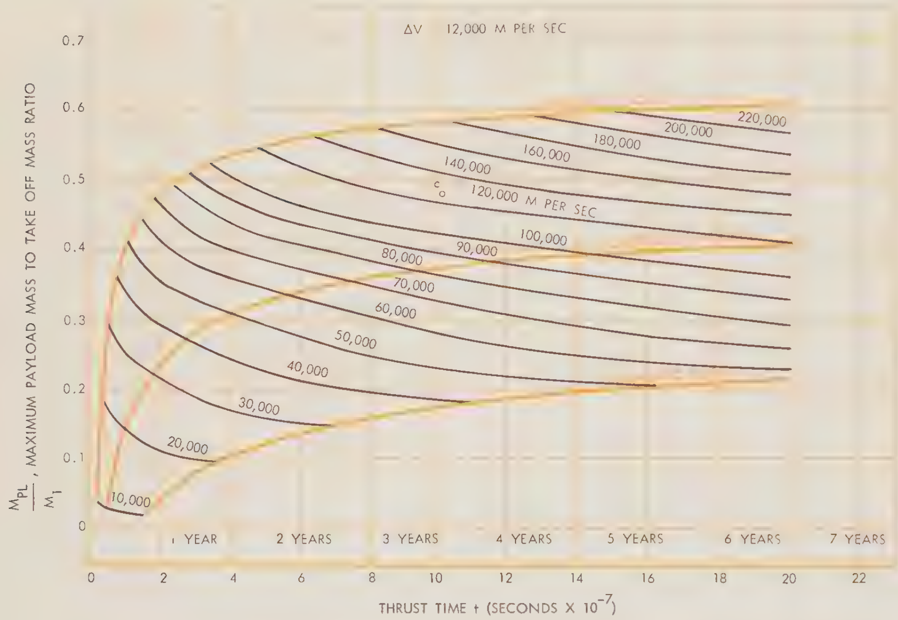
Discussed in this Appendix are the results of an analysis made by Allison research engineers on three basic electrically powered space rockets, referred to as the *optimistic rocket*, the *compromise rocket*, and the *pessimistic rocket*. The optimistic rocket was based on projected efficiency ratios. The compromise rocket was based on efficiency ratios which appeared possible at the time of the analysis. The pessimistic rocket was based on efficiency ratios which, at the time of the analysis, were not attainable.

The purpose of the analysis was to determine the optimum exhaust velocity c_o required for various mission velocities ΔV. The optimum exhaust velocity was

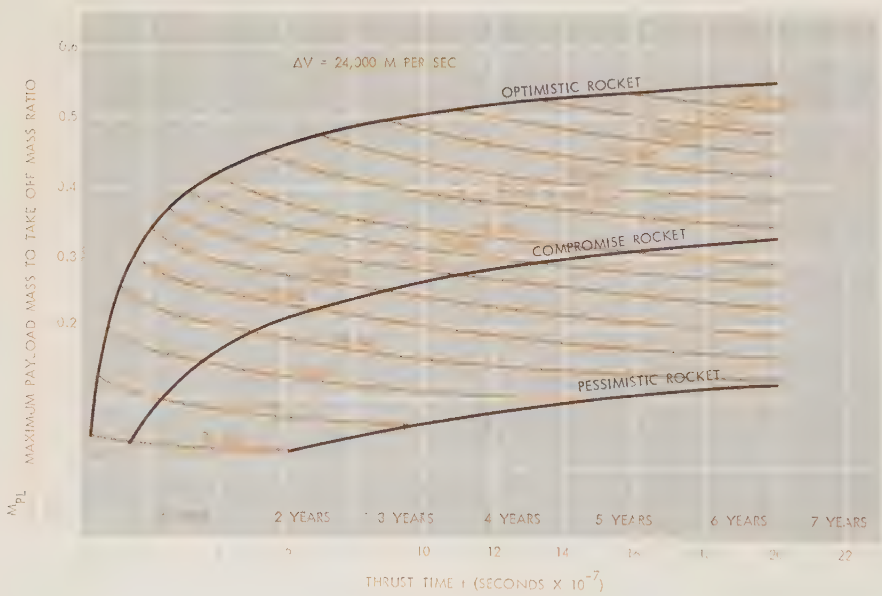
1A



2A



3A



determined for mission velocities of 3,500 m per sec (Fig. 1A), 12,000 m per sec (Fig. 2A), and 24,000 m per sec (Fig. 3A).

The mission velocity of 3,500 m per sec corresponded approximately to a simple escape maneuver from an Earth orbit. The 12,000 m per sec mission velocity corresponded to a one-way flight from an Earth orbit to an orbit of Mars or Venus. The mission velocity of 24,000 m per sec corresponded approximately to a round trip between Earth and orbits of Mars or Venus. It was realized that the actual mission velocities for the three flights would depend on rocket accelerations and thrust time[6] as well as the orbits chosen and trajectory corrections in flight. The exact velocities, however, would not differ greatly from the mission velocities chosen for the analysis.

For the simple escape maneuver from an Earth orbit (Fig. 1A), exhaust velocities lower than 5,000 m per sec (specific impulse less than 500 sec) were not considered, since at this point higher payload to take off mass ratios can be achieved by non-electrical rocket engines.

The analysis (Fig. 2A, Fig. 3A) showed that optimum exhaust velocities less than 100,000 m per sec (specific impulse less than 10,000 sec) must be used by rockets feasible at that time on missions to Mars or Venus involving thrust times less than five years. For the round trip mission to Mars envisioned by Stuhlinger[2] (thrust time about two years), the optimum exhaust velocity of a rocket was about 59,000 m per sec. The optimum exhaust velocity of even the optimistic rocket was no greater than approximately 220,000 m per sec (specific impulse approximately 22,000 sec) for a five-year mission to Mars or Venus. For the pessimistic rocket, the optimum exhaust velocity was only 42,000 m per sec (specific impulse approximately 4,200 sec) for the same space mission.

In general, the three plots showed that if for some reason the exhaust velocity of an electrical rocket must be increased while still maintaining maximum payload mass to take off mass ratio, the necessary thrust time for a given mission also is increased. If mission time is limited, however, the optimum exhaust velocity and corresponding ratio of payload mass to take off mass would be completely determined by the efficiency ratios of the power generator, radiator, and structure used to support the payload.

Ultrasonic Cleaning: Its Theory and Some Factors Affecting its Application

By RICHARD W. THOMAS
Diesel Equipment
Division

A cleaning process which depends on cavitation for its effectiveness

High frequency, or ultrasonic, sound waves have been used as the operating medium for equipment found in such diversified applications as detecting interior flaws within solid bodies, particle precipitation (smoke, fumes, fog), submarine detection, and methods of machining, welding, and soldering. One of the latest applications of ultrasonics is in the field of industrial cleaning. High frequency sound waves, transmitted throughout a cleaning fluid, cause cavitation to take place in the liquid. Microscopic bubbles form and collapse thousands of times a second to produce a powerful scrubbing action which pulls soil particles from the part being cleaned. With proper application, ultrasonic cleaning has proven advantageous in manufacturing and processing operations requiring parts or intricate assemblies to have a high degree of cleanness.

During the past few years, the trend toward more rigid requirements in the manufacture of close tolerance parts has included the need to have such parts possess a high degree of cleanness*. The removal of both soluble and insoluble soils by established cleaning methods such as power washers, sprays, dips, and agitation cleaning systems has not always provided the exacting cleaning requirements desired. A more effective cleaning method was sought which also would be able to handle intricate parts or assemblies of complex design too delicate to clean by the established methods. The result has been the development of ultrasonic cleaning—a method which has become in a relatively short span of time an important tool in the field of industrial cleaning.

Up to a few years ago, the ultrasonic cleaning method was mostly a laboratory curiosity. Applied research, however, plus experience gained in its application and improvements made in some of its major components have raised it to its present state as a technically important and economically feasible production process.

For example, included in the products manufactured by Diesel Equipment Division are certain close-tolerance mating components which require a high degree of cleanness prior to their assembly. The ultrasonic cleaning method was explored as a possible way to improve the over-all efficiency of cleaning procedures required for such parts. After extensive laboratory work indicated that ultrasonic cleaning could be effective, the method was applied to production cleaning operations (Fig. 1).

**Cleanness* is preferred terminology in ultrasonic cleaning work.

High Frequency Sound Waves Transmitted in Liquid Cleaner

An ultrasonic cleaning unit has three components: (a) a generator or motor alternator, (b) a transducer, and (c) a cleaning tank. The generator produces a-c electrical energy. The transducer, coupled to the generator by a co-axial conductor, converts the electrical impulses into high frequency sound waves which, in most cases, are above the frequency detectable by the human ear. The cleaning tank, usually made of stainless steel, holds the required liquid cleaning agent into which the transducer transmits its sound energy.

Ultrasonic sound can be divided into a low frequency and a high frequency range. With 50,000 cps taken as an arbitrary division point, the low frequency range extends from 16,000 to 50,000 cps while the high frequency range includes everything above 50,000 cps. In general, the combined spectrum of both low and high ultrasonic frequencies includes the range from 16 kilocycles per sec to several million cps.

The principal differences between high and low frequency ultrasonic irradiation are:

- Particle displacement for a given amount of power is inversely proportional to frequency and is much larger at low frequency than at high frequency

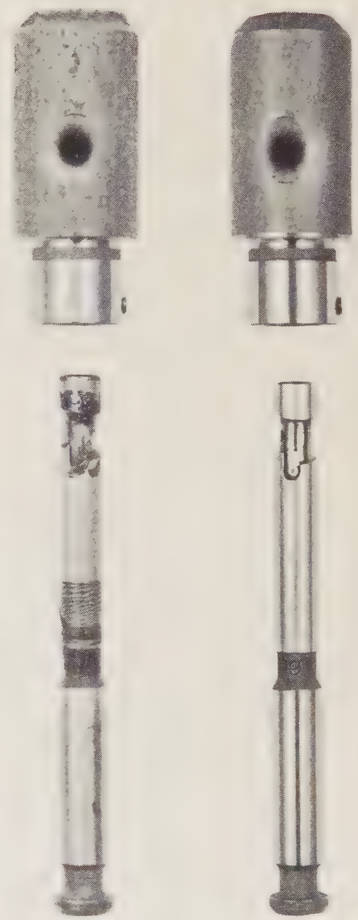


Fig. 1—These photographs of the plunger and bushing components of a Diesel engine fuel injector manufactured by Diesel Equipment Division illustrate typical results of ultrasonic cleaning applied to parts having highly finished surfaces. The photograph at the left shows a plunger and bushing contaminated with buffing, lapping, and grinding residues prior to cleaning. The photograph at the right shows the same components after ultrasonic cleaning, rinsing, and drying.

- The wave length of sound in any medium, such as a liquid, also is inversely proportional to frequency.

These two differences, when applied to the ultrasonic cleaning process, produce entirely different effects.

Sound cannot be transmitted in a vacuum and must have a medium to carry it whether that medium be a solid, liquid, or gas. The velocity of sound in most liquids is approximately 1.5×10^5 cm per sec. This means that at a frequency of 20 kc per sec the length of the sound wave in a liquid is 7.5 cm. At higher frequencies, such as 400 kc per sec, the wave length is reduced to 0.375 cm. The different wave lengths produced at various frequencies in the liquid cleaning agents used in ultrasonic cleaning give rise to some problems, which will be discussed.

Ultrasonic Cleaning Method Depends on Cavitation

The designer of high speed hydraulic machinery tries to avoid, if possible, the phenomenon of cavitation because of the undesirable effects it produces, such as serious vibration problems and decreased efficiency. The designer of an ultrasonic cleaning unit, however, relies on cavitation to produce the agitating action which removes soil from the part being cleaned.

Collapsing Bubbles Produce Scrubbing Action

Cavitation occurs during ultrasonic cleaning because the alternating pressure transmitted throughout the cleaning agent by the transducers is sufficient to lower the instantaneous pressure at any point in the liquid below atmospheric pressure. Since most liquids are inelastic and have no tensile strength, negative pressures cannot be sustained. The superimposed alternating pressure produced by the vibrating transducer alternates about the ambient pressure of 14.7 psi and can achieve very high values.

The alternating pressure is dependent only on the amount of power put into the transducer and irradiated into the liquid medium. When the magnitude of the maximum alternating pressure exceeds the static ambient value, a negative pressure is produced in the liquid on negative half cycles. When negative pressures are produced in a liquid by the passage of sound waves propagated by the vibrating action of the transducers, ruptures are created throughout the volume of the liquid in the form of highly stressed microscopic bubbles, or cavities. These bubbles, when first formed, have a negative pressure, but gradually develop a very low pressure —namely, the vapor pressure of the liquid in which cavitation is taking place.

The vibrating action of the transducer produces alternating plus and minus pressure waves in the liquid. The minute bubbles are produced during the minus pressure wave and are destroyed by internal collapse, or implosion, during the following plus pressure wave.

The rapid collapse of the bubbles thousands of times a second in close proximity to the dirt particles produces a scrubbing action which pulls the soil particles from the part and disperses them into the liquid cleaning agent. Because cavitation is produced throughout the entire volume of the liquid in the cleaning tank, the scrubbing action is carried into very small blind holes, crevices, and grooves as long as these areas are free of entrapped air.

Cavitation More Intense at Low Frequencies

For a given power input, cavitation is more intense at low frequencies than at high frequencies because particle displacement for a given power input is inversely proportional to frequency. At constant power, the actual physical movement of a given molecule of liquid will be 20 times greater at 20 kc per sec than at 400 kc per sec. At 20 kc per sec, for example, the bubbles form and collapse at the rate of 20,000 times per second. As they collapse, instantaneous bursts of energy are released in the form of heat which may reach temperatures of several thousand degrees Fahrenheit, accompanied by pressures of 30,000 to 50,000 psi, or higher. Sufficient time must be allowed during the minus pressure wave to allow the bubbles to grow to a usable size before collapsing. If frequencies are too high, the bubbles cannot reach a size which will supply any appreciable energy upon collapse.

To form the bubbles in a liquid some of the acoustical energy must be expended to create interfaces. Since more power is required at high frequency to bring about bubble formation, more energy is expended. On collapse of the bubble, therefore, heat energy is released

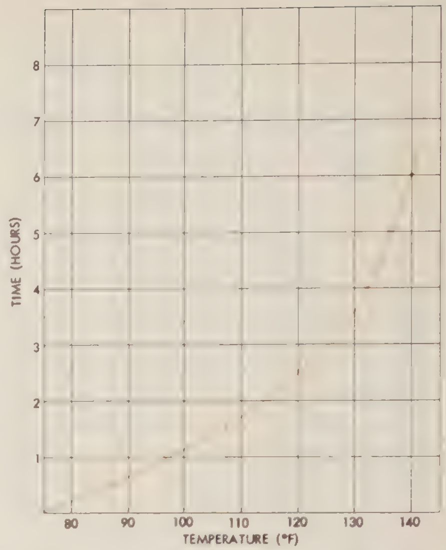


Fig. 2—The cavitation occurring during ultrasonic cleaning is a source of heat energy. This curve shows the temperature rise of a water-liquid nonionic detergent solution subjected to a frequency of 20 kc per sec for a period of six hours. The average power input was from three to nine watts per sq. in. of transducer surface. The capacity of the cleaning tank was 5 gallons.

in greater amounts than accompanies the bubble collapse at low frequency.

The collapsing bubbles also act as local sonic wave generators, with widely varying sonic frequencies. To obtain cavitation, however, high energy of sufficient density must be transmitted throughout the liquid. Below the required density level there is no cavitation. Above it, cavitation becomes more violent and cleaning more effective as intensity is increased. The density level of cavitation increases as the frequency increases, but not linearly. The upward curve is gentle between 110 and 18,000 cps, then it starts to rise sharply until at one megacycle it is nearly vertical.

Generated Heat a Useful By-Product

In general, the higher the acoustical frequency, the higher the heat level attained by the fluid medium. After several hours of continuous operation of an ultrasonic tank, this heat gain reaches a plateau (Fig. 2). In many cases the heat which is generated is a useful by-product of the cavitational phenomenon, since most cleaning agents work more effectively at warm temperatures.

There also is an optimum cavitation point for most liquid cleaning agents which is somewhat dependent upon

temperature. For aqueous base cleaning agents this is usually between 130°F and 170°F. Chlorinated solvents usually perform best at room temperature, but unfortunately their heat gain is quite rapid (Fig. 3). The result is that those cleaning agents with low boiling points soon cease to cavitate, unless their temperature is kept below the critical point by heat exchangers.

Frequencies Used for Removing Soils Vary

There is still some question as to the best frequency to use for ultrasonic cleaning. Various soils and contaminants may each have optimum frequencies at which they are most effectively removed. A specific soil, for example, may best be removed from a small, lightweight part at one frequency but may require a different frequency if it must be removed from a heavy part. Also, further frequency variations may be necessary depending on whether or not the soil is on the surface, in a through hole, or in a blind cavity. Present day ultrasonic cleaning units have frequencies ranging from 19 to 400 kc per sec, with some units providing multiple frequencies at both the high and low levels.

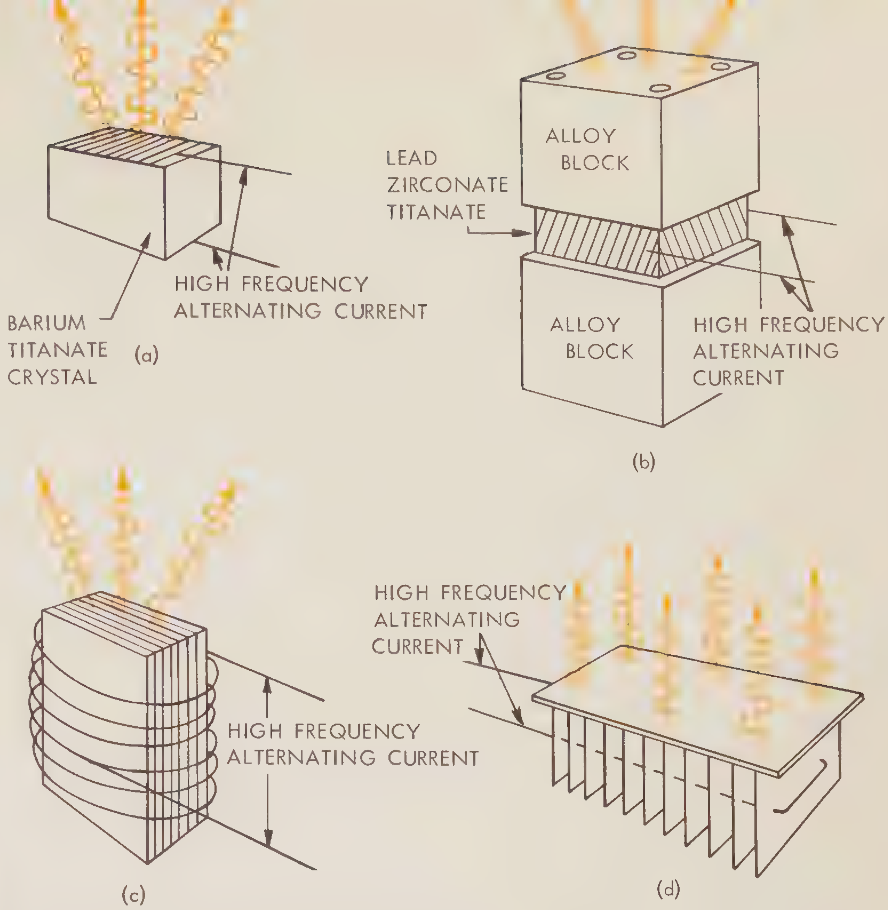


Fig. 4—The transducers used in ultrasonic cleaning work to transform electrical energy to mechanical energy in the form of sound waves are made of either electrostrictive or magnetostrictive materials. The barium titanate crystal (a) is used in the majority of electrostrictive transducers. Another type of electrostrictive transducer makes use of thin sections of lead zirconate titanate sandwiched between blocks of special alloys (b). Nickel is most commonly used as the magnetostrictive transducer material. Two commonly used types of magnetostrictive transducers are the nickel strip type (c) and the nickel space laminate type (d).

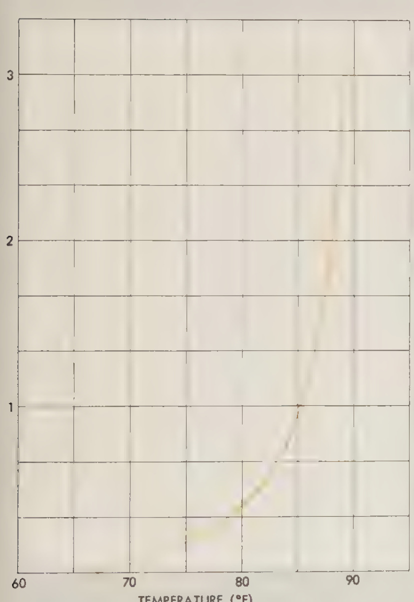


Fig. 3—Chlorinated solvents used as ultrasonic cleaning agents work best at room temperatures, but exhibit a rapid heat gain after continuous operation of a cleaning tank. This curve shows the temperature rise of methylene chloride irradiated at 40 kc per sec for three hours in a tank having a capacity of 5 gallons. The average power input was 2.2 watts per sq in. of transducer surface. The rise in temperature occurred even though a single-loop cooling coil, carrying water at a temperature of 58°F, was placed in the solvent.

Cavitation Causes Transducer Erosion

The cavitation occurring with ultrasonic cleaning causes erosion to take place, but not with equal force, on all metal surfaces of the transducer in contact with the sonic bath. Usually, the harder the metal, the slower the eroding action. This fact is used as one means for obtaining information on energy densities within cleaning solutions. It is accomplished by measuring the amount of metal removed from lead test plates over predetermined periods of time.

The greatest rate of erosion occurs where the energy density is greatest. This is at the portion of the tank surface immediately above the transducers.

Two Different Materials Used for Transducers

The transducers used in ultrasonic cleaning units are made of either electrostrictive or magnetostrictive materials. The two basic electrostrictive materials used are pure barium titanate and lead zirconate titanate. The magnetostrictive material most commonly used is nickel.

Electrostrictive Transducers

An electrostrictive material expands and contracts in response to voltage alternations. The frequency at which a material having this property vibrates is primarily determined by its mass, as defined by its physical dimensions.

Some of the electrostrictive transducers used in the earliest ultrasonic cleaning units were made of quartz crystals. For approximately the past eight years, however, the barium titanate crystal has been used in the majority of electrostrictive transducers (Fig. 4a).

Another type of electrostrictive material used is lead zirconate titanate. Because it is difficult and expensive to manufacture in heavy sections, transducer designs have been developed which use thin sections of the ceramic material sandwiched between blocks of special alloys which are positioned, machined, and dimensioned to produce a transducer which vibrates at a frequency between 20 and 25 kc per sec (Fig. 4b).

Barium titanate crystals have some limitations. If the crystals are made large enough to take advantage of frequencies below 40 kc per sec, their physical strength is not great enough to withstand the dimensional change and they shatter. The crystal is a relatively poor heat transfer medium. Although its Curie point* is approximately 240°F, cleaning tanks equipped with some types of barium titanate transducers cannot be operated with liquid temperatures above 160°F. Finally, the material is limited to a maximum power input of five watts per square in. due to its fragility.

The sandwich type of transducer using thin sections of lead zirconate titanate, on the other hand, has certain advantages. This type of transducer has a 90 per cent efficiency as compared to 70 per cent for barium titanate. The sandwich type of transducer has a higher Curie point which allows bath temperatures up to 200°F. Only 10 per cent of its input energy is converted to heat, as compared to 30 per cent for barium titanate. Finally, the transducer has the ability to accept and convert higher electrical input to sonic energy.

Magnetostrictive Transducers

A magnetostrictive material is one which experiences a change in length when subjected to a magnetic field. This property is exhibited by the ferromagnetic metals iron, nickel, and cobalt and their alloys. The change in length is, of course, very small. For nickel, which is used in most magnetostrictive transducers, the change in length is approximately 30 parts in one million. At resonance, however, the change in length, which is limited by fatigue properties, may be as high as one part in one thousand.

When first introduced, magnetostrictive transducers cost more than electrostrictive transducers. Of more concern, perhaps, was the fact that magnetostrictive transducers were only 50 per cent efficient, because of eddy currents and hysteresis losses, and that power requirements were anywhere from 100 to 200 per cent greater than for electrostrictive transducers. In addition, magnetostrictive transducers generated considerable heat (part of the reason for the high power input) and required additional plumbing for a cooling system. They did have one distinct advantage over electrostrictive transducers—they could tolerate bath temperatures of approximately 250°F. Recent developments in magnetostrictive transducer design have made it possible for them to compete favorably, both in cost and operating characteristics, with electrostrictive transducers.

The magnetostrictive transducers used today are of two basic types: the nickel strip type (Fig. 4c) and the nickel space laminate type (Fig. 4d). which converts the entire bottom of a cleaning tank into a uniform radiating surface. These two transducers work best at frequencies below 40 kc per sec and, therefore, take advantage of the fact that cleaning efficiency increases as the frequency values approach 10 to 16 kc per sec. They also will withstand heavy power levels without damage. In addition, they have high Curie points and, although they cease to function properly if bath temperatures exceed 300°F, they will not be damaged permanently.

The nickel space laminate type of transducer can handle power densities up to 30 watts per square in. of transducer surface. It should be mentioned, however, that there is a limitation to the amount of power, in watts, that can be transmitted into an ultrasonic cleaning bath. Any given liquid is capable of accepting a fixed amount of power from a plain radiating surface. If this fixed input is exceeded, surface cavitation occurs in the immediate vicinity of the radiating surface. This tends to form a barrier to the energy being transmitted from the transducer. This barrier prevents cavitation of much of the rest of the bath and radically reduces cleaning efficiency.

Cleaning Agents Must be Carefully Selected

The speed and efficiency of ultrasonic cleaning is dependent on two factors: (a) the cleaning agent must have a certain degree of compatibility with the soil to be removed and (b) stable standing sonic waves must be maintained in the liquid for the most efficient cleaning.

Stable standing sonic waves exist in a liquid irradiated from the bottom of a tank when the surface of the liquid is at a node—that is, at one of the odd quarter wave lengths from the antinode or radiating surface. When a transducer is located on one side of a tank, therefore, the opposite side of the tank must be located at a node to produce stable standing waves.

All liquids cannot support an efficient cleaning action, since the phenomenon of cavitation requires a cleaning fluid to have:

- A relatively low vapor pressure at the working temperature of the bath which will remain thin bodied and non-viscous at that temperature
- A density approximately equal to that of water or a little higher.

Many of the cleaning agents used in various industrial types of power washing equipment have been applied to ultrasonic cleaning. Other heavy duty detergents, wetting agents, and alkaline cleaners not adaptable to power washers also have proven very useful as ultrasonic cleaning agents. Such agents, by careful formulation, can be compounded to provide cleaning baths which will:

- Lower the surface tension of water to allow easy bubble formation with less expenditure of energy and, consequently, less loss of energy through heat transfer
- Lower the inter-facial tensions existing between the dirt particles and the part being cleaned
- Permit faster penetration of the liquid into the masses or layers of dirt as well as small holes, grooves, and cracks
- Allow, in some cases, chemical reaction with the soils, either dissolving them directly or forming soluble soaps (with some types of dirt) which are more readily solubilized
- Aid in dispersing the dirt removed and in resisting its tendency to redeposit on the parts being cleaned.

Proper Rinsing Important

Non-soluble soils removed from parts being cleaned remain in suspension within the fluid medium. To some extent, the

*The Curie point is the temperature at which depolarization occurs causing the crystal to stop functioning as a transducer.

non-soluble particles may be redeposited on the parts when the bath reaches a certain degree of contamination. This redeposition is more likely to occur within a one-half in. deep area immediately below the surface of the bath since cavitation, or at least the cleaning efficiency, appears to fall off within this space. In heavily contaminated baths, therefore, some redeposition of soil may occur as the parts are removed from the bath. This is not necessarily detrimental to the ultimate cleanness of the part, providing proper rinsing procedures follow the initial cleaning.

If a filtering system is incorporated with the cleaning bath, the rinse cycle is not quite so critical. Since organic solvents are usually easier to filter than aqueous solutions, ultrasonic cleaning units using organic solvents often use a continuous filtration of the cleaning bath.

Diesel Equipment Division has found it more economical when using detergent type baths to follow the initial cleaning with a dip rinse in an overflow bath and then return to a sonic rinse. With this procedure, the contamination level of the cleaning tank can reach a high level without affecting the cleanness of the part. When possible, recovery of the cleaning solution is accomplished by separate filtration after it is removed from the tank.

An important fact must be kept in mind regarding filtering or cascade overflow purification systems. All fluids contain certain amounts of dissolved gases which, in most cases, are removed rather rapidly from the solution as soon as cavitation begins. Evidence of this de-gassing action are the streams of bubbles which rise to the surface during the first few minutes of ultrasonic irradiation. These bubbles should not be confused with other visible swirls, streams, or eddy currents usually found in baths being irradiated with fairly high power densities. The bubbles formed during de-gassing act as an energy sink which may rob a transducer of two-thirds of its available energy. This is why all operating instruction manuals for ultrasonic cleaning units call for a warm up period of anywhere from a few seconds to several minutes, depending upon the cleaning agent being used.

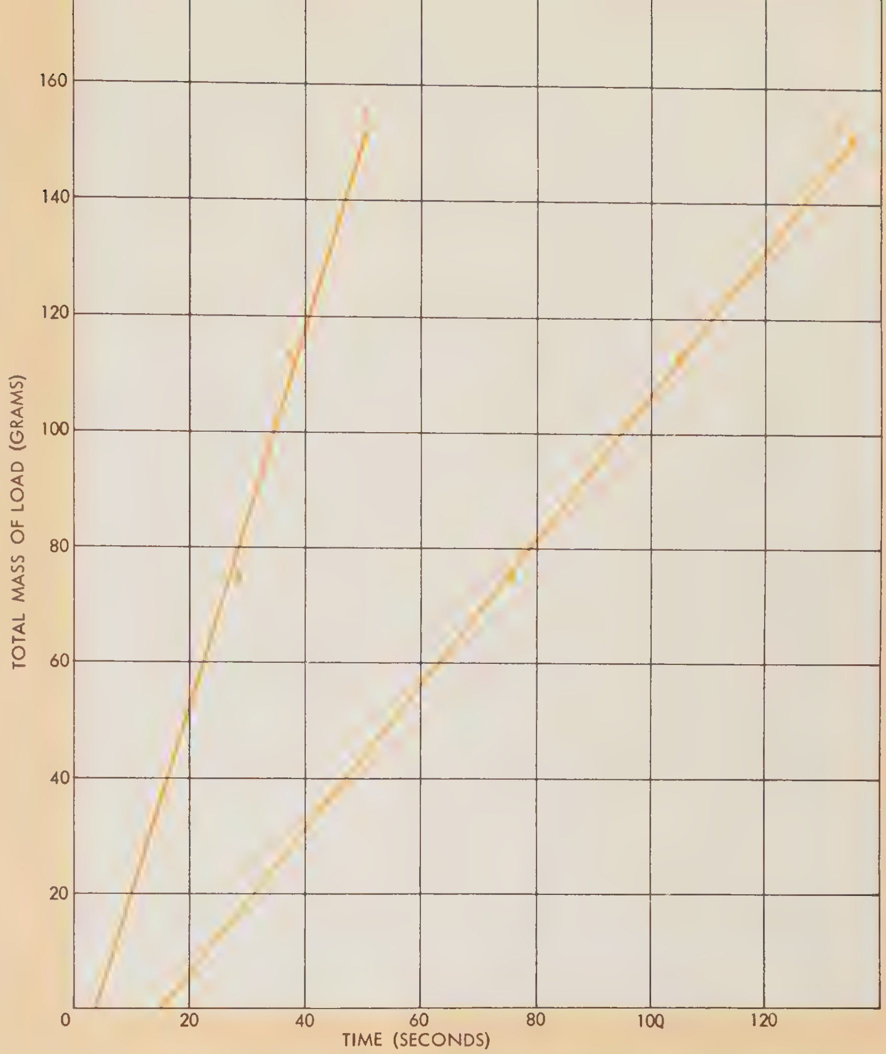


Fig. 5—This is a graphic representation of the effect of mass upon cleaning time at two levels of acoustical power input to the transducers. Curve *A* was obtained from an input of 60 watts which provided 1.9 watts per sq in. of transducer surface. Curve *B* was obtained from an input of 300 watts providing 2.2 watts per sq. in. of transducer surface. The volume of the ultrasonic tanks used for gathering the data was ½ gal for curve *A* and 5 gal for curve *B*. These curves are only applicable to the same bath composition, volume, and temperature at which the tests were conducted. With increased power input, the slope of the curve inclines more toward the vertical, indicating that greater levels of power input will compensate for larger parts or number of parts with less increase in cleaning time.

Volume and Depth of Bath Affect Cleaning Efficiency

The introduction of fresh non-degassed fluids into an operating cleaning bath at too high a flow rate may destroy its cleaning efficiency. A flow rate of more than one or two per cent of the volume of the tank per minute is incompatible with efficient cavitation. This does not apply to chlorinated solvents which, due to their high specific gravities, do not readily absorb gas. The volume of the transmitting fluid and its depth, temperature, viscosity, vapor pressure, and degree of contamination are all interrelated. Because of variation in these factors, some fluids do not cavitate as readily as others and consequently do not clean with equal degrees of efficiency or effectiveness.

Another consideration regarding volume and depth of the cleaning bath is the possibility of reflection. If the surface of the bath is too close to the diaphragm of the transducer, the energy wave will be reflected back to the diaphragm (tank bottom) with little attenuation. This action greatly accelerates erosion of the tank bottom, with its attendant maintenance problems. Data have been collected and published to show that the transmission of ultrasonic energy from transducer face to cleaning solution is more efficient when the face of the diaphragm is smooth and free from irregularities.

Effective cleaning can be done anywhere within the fluid medium, providing the power input is sufficient to produce the level of activity necessary for each particular cleaning operation or application. Experience has shown that processing time may be shortened somewhat if

the work is located within the bath at some antinodal point—some multiple of a half-wave length above the tank bottom. In practice, however, it is rarely possible to achieve such critical positioning. Fortunately, the positioning requirement can be almost completely compensated for by moving the parts through the cleaning medium in both a vertical and horizontal plane. This motion brings the parts into areas of both high and low cavitational energy. Also, physical movement of the fluid is provided over and around the parts. This aids in washing away loosened dirt.

Containers Should Not be Overloaded

The containers or racks used to suspend parts in an ultrasonic cleaning bath should: (a) cause minimum interference to the standing sonic waves and (b) provide high accessibility of the parts to the ultrasonic irradiation. The best practice is to suspend the parts to be cleaned from a rack, but small size parts having a cylindrical, cubical, or spherical shape are not adaptable to racking. In these cases, it is more convenient to use containers such as baskets or beakers.

Considerable work has been done to show that when baskets are used they should be constructed of either thin (0.010 to 0.020 in.) metal sheets, or of woven metal cloth whose openings are greater than ¼ in. or less than 200 mesh. Screening sizes falling between these size limitations cause an undesirable scattering of much of the acoustical energy reaching the container. This same effect is produced by closely packing large numbers of small parts into a container. The scattering effect is further complicated by the fact that the vibrating action caused by the acoustical energy creates an even tighter compaction of the parts. Proper load size is more critical at low frequency than at frequencies above 100 kc per sec. The higher frequencies have a greater power to penetrate deep recesses, and the small spaces between very small parts can be considered as recesses.

Early exploratory work by Diesel Equipment in applying the ultrasonic cleaning method indicated that the cleaning efficiency of a sonic bath was specifically related (a) to the mass of the part, or load of parts, being cleaned, and (b) to the time required to achieve the desired degree of cleanness. Curves were plotted of this effect (Fig. 5) using data obtained from ultrasonic units having power input levels of 60 watts and 300 watts. The curves provided a basis for estimating the cleaning time required for larger loads, commensurate with the total acoustical energy available.

The experience of Diesel Equipment with the ultrasonic cleaning of fairly large numbers of small parts has been in the removal of a protective oil film along with minute residual particles from 7/32-in. and ¼-in. diameter steel balls used as valves in hydraulic valve lifters. At present, loads of from 7,500 to 10,000 pieces are being satisfactorily cleaned using a constantly filtering bath of trichloroethylene irradiated with a high frequency of 390 kc per sec. Immersion time is approximately three minutes.

The principal drawback of high frequency cleaning, when applied to parts considerably larger than the steel spheres, is a shadowing effect which accompanies irradiations of short wave length. Low frequencies or long wave vibrations will tend to reach around corners of obstacles by diffraction and tend to eliminate shadowing. High frequency irradiation, however, causes many problems produced by zones and shadowing which result in some exceptionally clean areas as well as areas quite unclean on the same part.

In some instances, the positioning of parts in relation to the standing waves is of considerable importance. Cylindrical parts are usually arranged so their longitudinal axes are parallel to the standing waves, especially if the cylindrical part has a central bore through its axis.

Summary

In a relatively short span of time, the ultrasonic cleaning method has developed into an important production process for meeting exacting cleaning requirements. To be most effective, however, the following factors should be considered in regard to an ultrasonic cleaning operation:

- The type and solubility of soil(s) to be removed
- The size, shape, and weight of the parts to be cleaned
- The number of parts to be cleaned
- The frequency to use for effective soil removal
- The relationship between the mass of the part, the sound energy level of the ultrasonic bath, and the cleaning cycle time
- The positioning of parts in the container to obtain the best cleaning efficiency.

Although ultrasonic cleaning will not replace the other established industrial cleaning methods, it does have certain advantages such as the ability to

- Clean inaccessible areas
- Remove soil films having strongly adherent forces
- Reduce cleaning time
- Remove soils in the 10-micron and sub-micron range
- Remove soils from intact intricate mechanisms or from parts and assemblies too delicate to clean by other methods.

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Applying Instrumentation to the Evaluation of Noise in Automotive Accessory Motors

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Delco Appliance Division

More information about the principal problem: brush and commutator noise

Automotive accessory motors, particularly heater-defroster motors, are commutating, direct-current motors operating in the speed range of 2,000 rpm to 4,000 rpm. In such motors, brush noise, magnetic noise, and vibration and mechanical noises are inherent and must be controlled to have an acceptable product. In the past, such noises have been evaluated in a general way by personal judgment or cumbersome and questionable attempts to use so-called noise meters. An improved method of noise evaluation, using available instrumentation, has been developed and applied at Delco Appliance Division. This method translates noise into a finite numerical value.

ELECTRIC motors for automotive accessories are small, commutating, direct-current motors. They are used in such applications as automotive heaters and defrosters, windshield wipers, and actuators for seats, windows, and convertible tops. Motors of this type are manufactured in relatively large quantities to supply the needs of several automobile assembly plants. Delco Appliance Division often manufactures over one million motors a month for automotive accessories, and, in addition, builds motors for non-automotive applications and military equipment.

In accessory motors that operate at speeds ranging from 2,000 rpm to 4,000 rpm, such as heater and defroster motors, noise is inherent and must be controlled. The Noise and Vibration Laboratory at Delco Appliance has conducted studies to evaluate motor noise and to contribute to the over-all effort of its control.

How Motor Noises Are Caused

To evaluate noise in motors, the different sources of noise must be identified and understood. It is of interest, therefore, to review some of these sources before describing the laboratory investigations that were made.

The possible causes of accessory motor noise can be placed in two general classifications: electrical and mechanical. Many of the causes are interrelated but, for clarification, each one is considered separately.

Electrical Causes

Electrical causes of noise can be broken down into two categories:

(a) *Magnetic*—caused by a varying magnetic field acting upon various motor components. A variation in magnetic field is inherent in all d-c motors. The armature teeth cause variations as they enter the field. Commutation cannot be perfectly smooth and, as a result, variations occur which can cause noise. Skewing the armature, tapering the field poles, or increasing the number of poles or commutator bars are methods of reducing the effect of magnetic field variation. Maintaining mechanical tolerances which affect the magnetic fields also helps.

(b) *Brush arc*—caused by the sparking of the brushes on the commutator. Brush arc is affected by the brush position on the commutator in relation to the armature windings. To control this effect, any mechanical tolerances which would cause brush shift should be eliminated or sufficiently reduced. Arcing causes excess wear of the brushes and commutator bars which eventually causes noise. Arcing also is a sign that the magnetic variations are greater under such conditions. Indications are that the magnetic variations caused by brush shift and indicated by arcing are more troublesome than the noise caused by the brush spark.

Mechanical Causes

Mechanical causes of noise are numerous but, in all cases, the cause results in either an unbalanced condition or a vibration of some component. Each component of an accessory motor can be a cause of noise (Fig. 1). Typical components are:

- *Bearing*. An out-of-line, rough, dry, or loose bearing results in noise. The

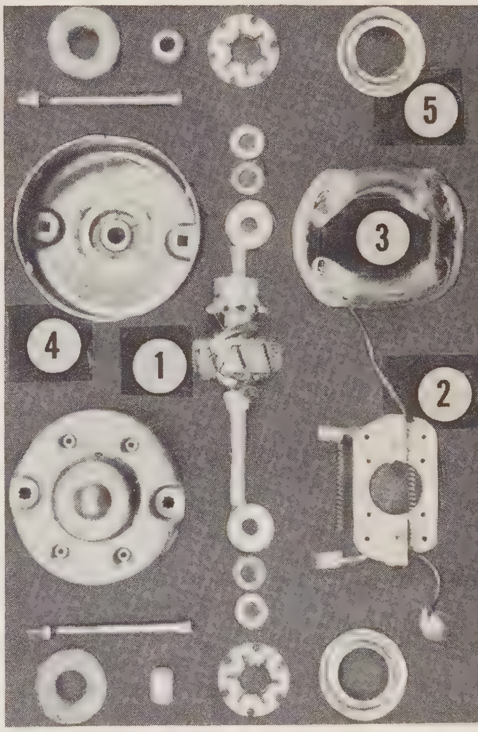


Fig. 1—The principal components of a typical automotive accessory motor investigated as possible sources of noise were: (1) armature assembly (commutator windings, stack, shaft, oil slingers, and washers); (2) brush holder assembly (brushes, brush springs, brush holders, brush plate, and terminal); (3) field stack and field windings; (4) end cases and tie bolts; and (5) bearings, bearing retainers, and oil wicks.

bearings are held in position by retainers and lined up by means of the case and field assemblies so the rule that each component can be a cause of noise holds true. There are some components which do not emit noise but produce a vibration which excites other components to produce noise. Bearing noise generally can be tied down to a frequency related to speed except where excess vibration results.

- *Armature*. Unbalance in an armature can be a cause of noise in accessory motors. The motor may not be radiating noise directly but, in an actual installation, unbalance of the armature will cause noise to be radiated and conducted throughout a car. The concentricity of the laminae and the uniformity of the windings contribute to the unbalance. The bearing surfaces can be damaged by the assembly of the armature components onto the shaft. A bent shaft can cause noise in extreme cases but in minor cases a bent shaft will generally result in vibration in an installation. Armature *strike* is caused by bearings being out of line (actually misalignment of the case), a loose wire in either the armature or field, or a projection of either field or armature lamina.

- *Washers*. Washer noise generally is caused by too much end play and the chucking of the armature in horizontally mounted motors. The noise is generally intermittent in nature and can be easily determined. The material used in making the washer is of great importance in determining the resultant noise. Burrs and bent washers and the order in which the various types of washer materials are placed on the armature shaft also are important factors in the amount of noise produced.

- *Slinger*. Slinger noise is combined with washer noise and generally emanates from similar causes. The material of the slinger can cause noise when struck by the washers. In extreme cases the slinger makes contact with the bearing face. The slinger also could damage the shaft journal surfaces, when it is pressed into position.

- *Brush holder assembly*. Its components affect the position of the brush on the commutator and also are possibly resonant and excited by the chatter of the brush. The line-up of the brush plate with the armature is determined by the accuracy of the end case and bearing retainer. Looseness of the brush plate due to improper riveting or a tilted condition of the plate can cause *bowing* of the brush plate which tends to counteract the brush clearance angle. Improper placement of rivet holes in the plate is usually the reason for this bowed condition. Bowing of the plate is undesirable because in compensating for the brush clearance angle, it causes the brush to present a larger area to the commutator. This produces more noise than when the brush is allowed to wear in gradually. The brush holder causes noise when associated with the spring and brush combination. The attachment of the brush holder to the plate presents a possible noise condition. A loose brush holder can cause a brush to ride on a corner instead of its designed contact surface, thus causing excessive noise.

- *Commutator*. The commutator is one of the principal producers of noise in a motor. *Bar-to-bar variation* has a direct effect on the amount of noise produced by the brush as does the width and roughness of slots and the eccentricity of the commutator with respect to the shaft. Bar-to-bar variation is the drop or rise between adjacent commutator bars.

- *Brush*. The accessory motor brush as a noise source is a common offender. Brush materials and brush making processes are very important in determining whether a brush is able to perform quietly or not. The design characteristics of the brush have a definite relationship to brush noise. Concave contact surfaces are more desirable from the standpoint of noise reduction than are flat surfaces. Brush spring pressure has a direct bearing on noise. Too little spring pressure causes excessive brush jump in the holder which results in undesirable noise. Excessive spring pressure results in a high pitched whine which can be considered objectionable.

Instrumentation Outranks Personal Opinion in Noise Measurement and Evaluation

Methods of noise evaluation previously used at Delco Appliance were based primarily on skilled personal judgment in the form of both individual and group opinions.

Evaluation of accessory motor noise during production was done by one of several inspectors who listened to a motor in operation. His opinion of the quietness determined whether the motor was accepted or rejected. Because of the differences in judgment among individual inspectors, a wide tolerance between accepted and rejected motors was possible.

Using this inspection method, a wide tolerance was possible for another reason. An inspector's ear became tuned to a certain noise level, and motors that deviated from this level were rejected. His evaluation was based on an average noise level for a production day. This level might be higher or lower on different days. Throughout a period of a week, therefore, the range of motor noise might

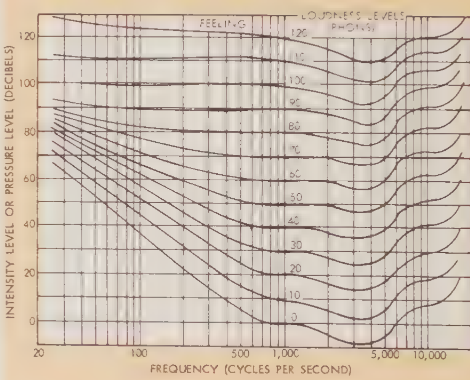


Fig. 2—This chart shows the sensitivity curves for the average ear subjected to pure tones in a free field and illustrates how ear sensitivity varies with frequency. To determine the loudness of a sound, the sound pressure level in decibels (2×10^{-4} microbar) of a 1,000-cps tone is adjusted so that the sound in question and the 1,000-cps tone appear to the ear to be of equal loudness. For example, a sound pressure level of 80 db at 50 cps is just as loud as a 1,000-cps tone adjusted to a 60 db sound pressure level. The loudness of the 50 cps sound, therefore, is expressed as 60 phons.

be very wide. A rejected motor on one day might be an accepted one on another day.

The differences in personal judgment make the establishment of standards for noise extremely difficult. Using a so-called master motor as a standard for noise is not satisfactory because the conditions within a motor can change during operation and thus change its noise characteristics.

In the laboratory, individual and group opinion similarly determined whether a motor was considered noisy. When design or processing changes were made, a reduction in noise sometimes could not be determined because of the lapse of time between the "before" and "after" tests. The observer could not remember what he had previously considered objectionable. In most cases, unless there was a very pronounced change in noise output, no real comparison could be made.

A step toward the use of instrumentation to improve the laboratory techniques was the application of a sound level meter and a sound analyzer. However, one result of their use was an indication that a frequency analysis would be necessary if motor noise was to be pin pointed to a direct cause. The use of a sound level meter by itself did not provide satisfactory results because sound level alone did not determine whether one motor was more noisy than another. Although two motors had identical indicated sound levels, observers would consider one more objectionable than the other. This condition would occur because a sound level meter indicates an over-all reading of noise level and does not differentiate with respect to frequency as does the human ear.

The use of the sound analyzer designed to function as part of the sound level meter enabled a breakdown of noise into components of frequency for more accurate noise evaluation. The effectiveness of the sound analyzer was limited, however. The time consumed in the manual sweeping of the frequency band presented results with varying degrees of accuracy, and repetition of noise measurement was not consistent. Microphone response of the sound level meter varied as a function of temperature which further reduced the possibility of accurate analysis. Generally speaking, for rapid and accurate analysis of frequency in accessory motor noise work, a greater stability of measurement was desired.

To improve the noise analysis techniques in the laboratory, work was directed toward the removal of personal judgment, both group and individual, while retaining correlation between what is heard by the ear and what is interpreted by artificial means.

Audio Frequency Spectrometer and Level Recorder Used

The next step in the application of instrumentation was the use of an audio frequency spectrometer and level recorder[1]. With this equipment, any noise can be graphically recorded as *sound pressure* (decibels *db*) versus *frequency* (cycles per sec) throughout the audio range of 40 to 16,000 cycles per sec in 27 third-octave bands. These graphical recordings of noise can be analyzed completely and accurately. The recordings of sound pressure versus frequency can be converted into units of loudness (expressed in units of *millisones*) for each band which can be dealt with individually. Also, the over-all loudness level (total of all loudness

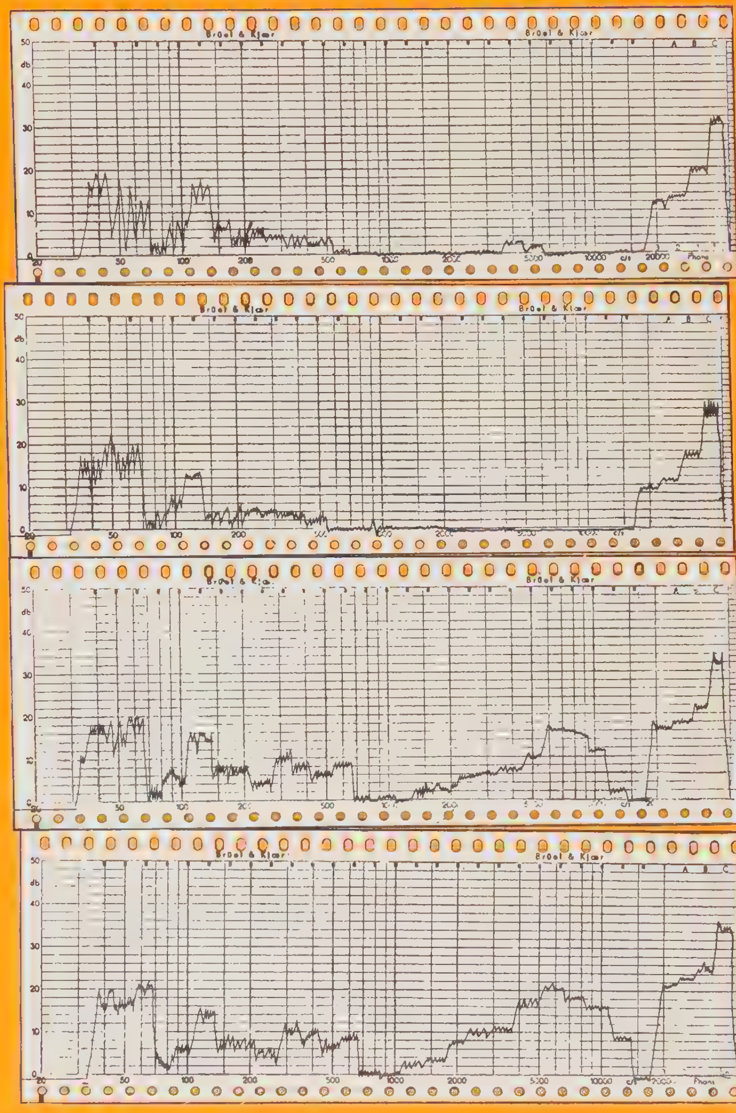


Fig. 3—An audio frequency spectrometer and level recorder is used in laboratory evaluation of motor noise. Shown here are typical spectrographs (amplitude versus frequency) produced in a particular test to evaluate motor noise when using brush materials from two different supplier sources. The first tape at the top shows a plot of the noise produced when the test fixture is operating without brushes. The second tape shows a plot of the ambient noise in a sound enclosure with the test fixture not operating. These are used for comparison after a test is made. The third and fourth tapes show the motor noise produced using brush materials from source A and source B. Loudness values for each frequency band and for over-all loudness are obtained using the charts in Fig. 4.

units at each frequency making up the noise) of the noise in question can be determined.

The over-all loudness level of a noise is expressed in *phons*. A loudness level of a pure tone in phons is numerically equal to the sound pressure level above 2 × 10^{-4} microbar of a 1,000-cps tone which sounds equally loud. For all practical purposes 2 × 10^{-4} microbar of sound pressure can be taken as the threshold of hearing. A phon unit is only a reference value using a 1,000-cps frequency as a criterion. For example, a sound pressure level of 80 db at a frequency of 50 cps is considered to be as loud as a 60 db sound pressure level at a reference of 1,000 cps. Thus, the loudness level of the 50-cps tone is considered as 60 phons (Fig. 2)[2]. It is only at a frequency of 1,000 cps that the phon scale and decibel scale become identical.

Phon values cannot be numerically added since they refer to loudness level and not to loudness alone. For this purpose the *sone* unit has been devised which enables the range of loudness to be divided into equal unit steps. One sone equals 1,000 millisones. A loudness of 1 millisone corresponds to the threshold of hearing. A loudness of 5 millisones is one-half as loud as a loudness of 10 millisones. The units of noise measurement used by the Delco Appliance Noise and Vibration Laboratory are based on these generally accepted standards of noise measurement.

Methods of Calculating Noise

Since loudness is a subjective quantity and cannot be measured exactly with any instrument, sound pressure level as graphically recorded must be corrected to the sensitivity curves of the average ear. The methods of noise calculation presently being employed in the laboratory are as follows:

(a) The spectrograph of sound pressure versus frequency of the noise being investigated is compared to a similar spectrograph of the ambient sound pressure (Fig. 3). If any frequency component on the graph of ambient noise is within 10 db of the corresponding frequency on the graph of the noise being investigated a correction must be made, as indicated by a *background correction* chart (Fig. 4).

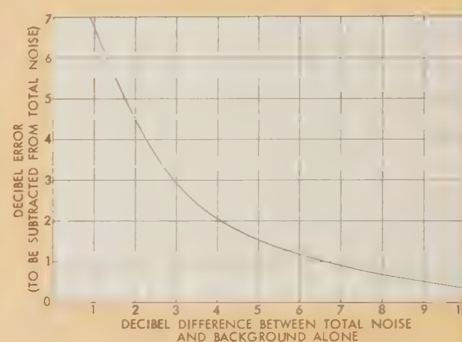
(b) From the corrected sound pressure levels, a loudness value in millisones is taken from a chart of sound pressure level versus loudness values for each frequency band (Fig. 4).

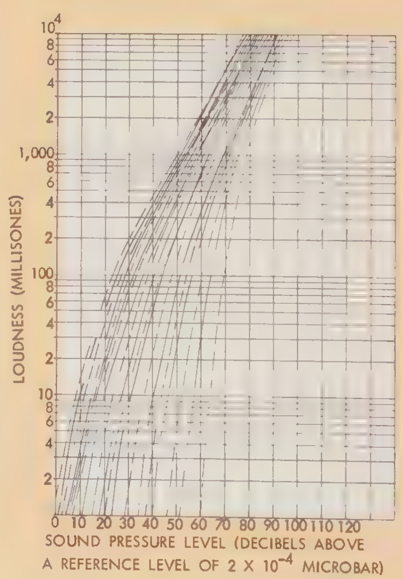
(c) The total of the loudness values in millisones of all frequency bands may be converted into an over-all loudness in phons (Fig. 4).

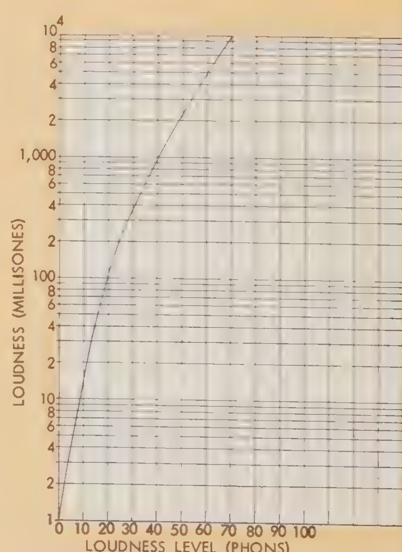
The laboratory method of summation of loudness values of frequency bands was supplemented by using another method which takes into account the effect of band spacing in noise measurements[3]. This method is considered more accurate when absolute noise values are desired.

The laboratory method of noise evaluation is adaptable to quality control work in manufacturing departments although some means must be used to reduce the time required to make noise measurements. Delco Appliance solved the problem with an instrument based on the same principles used in the laboratory except that it indicated instantly the loudness values in sones. This instrument, designed and built by GM Manufacturing Development, provides inspectors with a more positive means of measuring

Fig. 4—If a frequency component of the graph of ambient noise (Fig. 3) is within 10 db of the corresponding frequency of the noise being investigated, a correction must be made. From the *background correction* graph, shown at the right, a value for db error is subtracted from the total sound pressure level. Using the corrected level, a loudness value in millisones for each frequency band is read on the graph (bottom left). This is a plot of loudness in millisones for one-third octave filters as a function of sound pressure level in db above a reference level of 2 × 10^{-4} microbar. The total of the loudness values of all frequency bands is then converted into an over-all loudness value in phons by using the graph at the bottom right. This shows loudness in sones plotted as a function of loudness level in phons.







FREQUENCY (CYCLES PER SEC)	40	50	64	80	100	125	160	200	250	320	400	500	640	800	1,000	1,250	1,600	2,000	2,500	3,200	4,000	5,000	6,400	8,000	10,000	TOTAL
DECIBEL																										
CORRECTED DECIBEL															21	20	23	24	25	26	27	30	37	36		
MATERIAL A (MILLISONES)															38	33	50	75	100	125	121	134	43	1		720
DECIBEL																										
CORRECTED DECIBEL															21	23	24	28	31	32	32	38	41	38		
MATERIAL B (MILLISONES)															38	50	60	118	180	230	205	300	75	1		1,257

SAMPLE LOUDNESS CALCULATIONS		
EVALUATING: BRUSH MATERIALS A AND B		
MATERIAL A	720 MILLISONES	36.2 PHONS
MATERIAL B	1,257 MILLISONES	43.5 PHONS

Fig. 5—The calculations for noise values in laboratory evaluation work, described in Fig. 4, are summarized on a form as shown here. This spectrometer information is supplemented with tape recordings of the test to correlate the frequency composition of the noise with what the human ear hears. These two sources of information provide a reference for comparisons that can be made at a later time.

noise and, thereby, improves the inspection technique.

To eliminate interference from noise components other than those being investigated and to obtain accurate reproduction in noise tests, random noises must be eliminated. To achieve a consistent background level, a special enclosure is used which isolates the test setup from undesired outside interference. This enclosure is in two pieces with hinges at the back. The interior of the enclosure is approximately 15 in. square and is constructed of perforated sound board. The interior of the enclosure rests within an outer shell which has a reflective surface of sheet metal. The interior of the enclosure is connected to the outer shell through shock mounts and separated from it by three in. of glass fiber insulation. The outer shell is supported by shock mounts located at the four corners. A condenser microphone is suspended from the top of the enclosure and can be adjusted to any desired height. This type of enclosure enables accurate comparison to be maintained in any series of noise tests. Noise reduction as a result of changes made to a motor can be evaluated accurately even though there may be a prolonged time lapse between tests.

To facilitate a thorough analysis of noise and to preserve tests for future reference, magnetic tape recordings are used in conjunction with the audio frequency spectrometer. Tape recordings of actual motor applications may be made at any location. An analysis of the tape recording can be made by direct coupling of the tape recorder to the spectrometer. In this way a direct correlation is achieved between the noise heard and its frequency composition (Fig. 5).

Binaural tape recordings are made for jury evaluation of noise. The main advantage of binaural recordings is that spatial orientation is preserved. This condition of realism is gained by using two microphones approximately 8 in. apart (average distance between the human ears) and two recording heads. When played back the sound is picked up by individual earphones worn by the listeners corresponding to the sound as received by each microphone. There is no noticeable mixing of sound in the recording system.

Sources of Motor Noise Investigated in Laboratory

Another application of this method of measuring noise is in laboratory work investigating the sources of motor noise. At Delco Appliance, the main problem was how to identify the major source and how to separate it from other factors that cause noise and may be more easily corrected.

The first problem encountered was to find a method which would determine whether the principal source was mechanical, electrical, or a combination of both. This problem was approached by making comparison tests between an accessory motor running under its own power and the same motor rotated by an external means. The motor was placed in the test enclosure and run at a uniform speed. The noise produced by the motor was picked up by a condenser microphone suspended within the enclosure. This noise was recorded graphically with respect to frequency by means of the audio frequency spectrometer and sound level recorder.

For the second test, the conditions of speed, direction of rotation, and sound enclosure were the same. The accessory motor was driven from outside the enclosure by means of a shaft extending through the enclosure and connected to a separate, split-phase motor. (Speed variation was obtained by the use of a variable frequency supply.) Noise values were recorded in the same way as in the previous test.

A comparison of spectrographs of a motor operating under no-load and spectrographs of the same motor driven externally revealed that the frequency structure and amplitude of the noise produced under both conditions could be considered identical except for a magnitude increase at ten times the fundamental rotational frequency which occurred only when the motor was under power. This was termed *magnetic* and corresponded to the 10 armature conductors entering the magnetic field for each complete rotation of the armature shaft. From the results of many tests conducted in the same manner, it was concluded that the principal source of objectionable noise was primarily mechanical.

As a result of this conclusion, further investigation of accessory motor noise was directed toward the elimination of mechanical noise. The components of a motor were evaluated individually with the aid of a specially designed fixture (Fig. 6). This fixture enabled studies to be made of washer noise, bearing noise, and commutator and brush noise.

Studies of Brushes and Commutators Show Promise

Much of the laboratory work was devoted to investigation of brushes and commutators, which appeared to be the principal sources of noise. This included a study of brush materials in order to find a way to evaluate new materials before they might be used in production.

The frequency analysis method, described above, proved to be accurate in

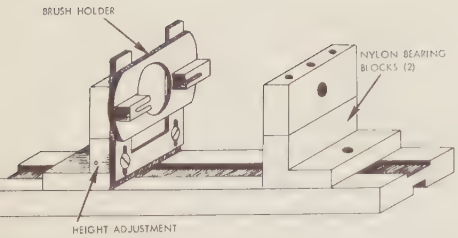


Fig. 6—This is a special fixture designed to hold an armature for laboratory evaluation of individual motor components. The armature is supported in nylon bearing blocks, which are removable and adjustable to provide conditions of misalignment. One bearing support has an adjustable frame on which a brush holder assembly is mounted for a test. Several types of motor noise can be studied using this single fixture.

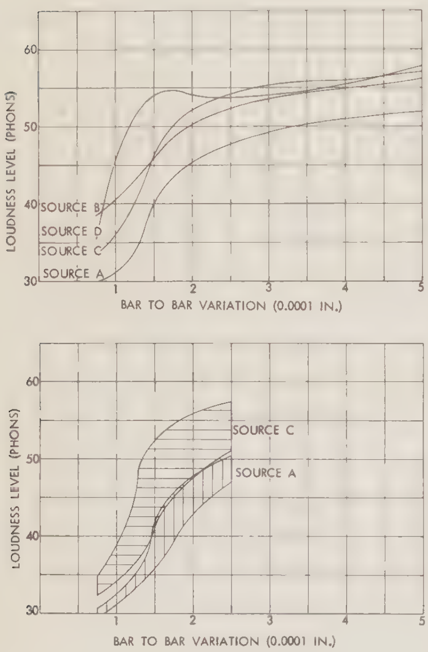


Fig. 7—These curves are plotted from data taken in studies of brush materials and bar-to-bar variation on commutators. The graph at the top shows the comparative loudness level of four different brush grades plotted against commutator bar-to-bar variation. These curves indicate the importance of keeping the bar-to-bar variation at a minimum to attain a lower loudness level. Also shown is the wide variation in loudness which is possible among different approved supplier sources of the brush material. The sources are rated *A, B, C, D,* in order of preference from a noise standpoint. The graph at the bottom shows how the loudness level varies among brushes from the same material source. (These ranges include test variations as well as material variations.) The data for the curves in these investigations were obtained by using different grades of brush materials alternately on commutators with known values of bar-to-bar variation. The curves represent an average from tests of four sets of each brush grade.

evaluating brush materials on a noise basis. For example, when the accessory motor noise was broken down by frequency analysis, observation showed that most of the audible noise occurred in the frequency range of 2,000 to 6,400 cycles per sec. An increase in motor speed increased the amplitude of the noise but no apparent frequency shift was observed.

An investigation of this condition indicated that the brush noise generated by an accessory motor consists of a series of impact noises. The sound pulses resulting are made up of frequencies in this 2,000 to 6,400-cps range. The noise structure does not depend on the number of times the impact occurs in a given period but on the frequency of the noise resulting from the impact. This indicates that the motor components are excited into a resonant condition by impact noises, possibly explaining why the frequency of the heater motor noise is not a direct function of motor speed.

The frequency analysis method also confirmed that objectionable motor noise is a function of commutator bar-to-bar variation and the general condition of the commutator. Motor noise is reduced when this variation is held to a minimum (Fig. 7).

Brush materials were evaluated by testing various combinations of brushes and commutators using brush materials from different supply sources. It was found that a fairly wide variation in the noise characteristics existed among materials from four approved supply sources (Fig. 7). This confirmed previous experience in the Inspection Department where motor rejections on the basis of over-all noise varied considerably with respect to the approved sources of brush materials. The laboratory tests enabled the sources to be graded *A*, *B*, *C*, and *D* in order of preference from the noise standpoint.

When a new sample brush material was submitted for evaluation, it was compared with the data on the existing materials. If the sample was found to be fairly close to source *A*, it was given further tests for performance and endurance. After passing these tests, a few brushes made of the sample material were used in actual motor production. Motors with the sample brushes were grouped with a representative number of motors using brushes from source *A* and all were inspected for noise. Any rejections from the entire group were then returned to the laboratory for evaluation with respect to the brush material.

Conclusions

The application of the frequency analysis method to investigating noise has been useful in evaluating automotive accessory motors. It has shown, for example, that brush noise is mechanical in nature and that the objectionable noise falls in the frequency range of 2,000 to 6,400 cps. It appears possible, therefore, to make production checks on motors either on a maximum noise level reading or on a "go" or "no go" basis by checking at 2,500 and 3,000 cycles per sec.

Evaluation of brush materials also has been aided. Data from the laboratory investigations can be used for comparing with new sample grades of materials. The selection of brush material from a noise standpoint seems to be a function of commutator bar-to-bar variation. Different grades of brush material react differently to varying degrees of bar-to-bar variation.

One of the principal problems encountered in investigations of this type is the presence of variables. Means of eliminating these variables are constantly being sought to enable faster and more accurate noise evaluation, not only of accessory motors, but also of other products that present similar problems. Work also is continuing in other areas that might affect motor noise, such as the finish of commutator bars and the mechanical resonance of motor components.

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Determining the Inventor and Related Matters

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Patent Section
Detroit Office

THE QUESTION of inventorship often arises as soon as an invention has been conceived and must be correctly answered before the patent application is filed. It is a problem which directly concerns the inventor, those who work with him, his employer or financial backer, his patent attorney, and the public. It is important to the public because the public is injured when an exclusive right to an invention is granted to a person other than the inventor. It is ethically important to the inventor and his patent attorney and the inventor's relationships with other people. It is also financially important to the inventor and persons or corporations to whom he may assign the invention or grant licenses under the patent, since any patent issuing in the name of one who is not the inventor, or omitting a joint inventor, is invalid unless the error can be and is corrected.

General Background

The Constitution empowers Congress to grant to inventors the exclusive right to their inventions for a limited period in return for a full disclosure of the invention to the public. Acting under this authority, Congress has passed statutes by which it has granted the right to obtain a patent, subject to certain limitations and conditions, to any person who has invented or discovered any process, machine, manufacture, composition of matter, or any improvement thereof. Plants, such as certain roses, can now also be patented.

The right to obtain a patent covering an invention is given only to an original inventor. To be an original inventor of something patentable, it is essential that the inventor himself be the one who conceived the patentable idea. One must also be the first inventor of the invention claimed in the patent, except under certain special circumstances which are beyond the scope of this article. In general, a person is entitled to a patent for his invention if it was original with him and he is the first one to perfect it and adapt it to use. He may be the first inventor if he was the first to conceive the idea and was diligent in adapting and perfecting his invention, even though he had the invention reduced to practice by another person, or another person who *later* conceived the idea managed to reduce the invention to practice before him. These and other facts may have to be legally established either during the prosecution of the application or in a patent suit. It is, therefore, necessary that complete records be kept of all facts surrounding the making of an invention.

How to Determine Who is the Inventor

In making this determination, the inventive features—what is new—should be established and then all available facts relating to the invention should be gathered, being careful to eliminate any wishful or biased thinking. These facts should include all sketches, drawings, photographs, and descriptions made pertaining to the invention, any notes made by the possible inventors or others closely connected with them and concerning the invention, and any oral and written directions given by one person to another about the invention. All meetings and conversations on the subject should be reconstructed, insofar as possible, together with the dates and places held, names of the persons present and their contributions thereto. Working models or samples made should be obtained, and any additions or changes made, when and by whom, should be noted.

An analysis of all available facts should then be made. The analysis should answer the following questions:

- Do informative materials and ideas given by one person to another contain the inventive features?
- Do the facts show that two or more persons worked in concert to conceive the inventive features?
- Do the facts show that the inventive features were actually conceived by a person receiving informative materials from another so that the person furnishing the informative materials is established as having no part in the conception of the inventive features?
- Do any or all of the changes made by other persons exceed what would be expected of the ordinarily skilled person familiar with the subject matter?
- Were any of the inventive ideas contained in disclosures made by others?
- Did samples made by one person follow the instructions of another and add nothing of an inventive nature thereto?

All personal relationship considerations must be eliminated so that a factual determination of inventorship can be made. The mere fact that one person supervises or employs another does not make either of them a sole inventor or join them together as joint inventors. The situation often arises under these circumstances wherein a problem to be solved is presented by one person to another. The person presenting the problem is not an inventor in any inventive solution to the problem. The person conceiving the inventive solution is the true inventor.

One who reduces sketches and ideas presented to him by another to a workable design, while using his ordinary skill as a mechanic or draftsman, for example, is not the inventor of the invention contained in the sketches and ideas. However, in producing the workable design, he may make an improvement to the invention embodied in the sketches and ideas. If the improvement is the result of his ingenuity beyond the ordinary skill of those familiar with the art, he has made an invention. The invention may be a joint one with the other person if they worked together to arrive at the improved result, or a sole invention if the improvement was conceived by him alone and rises to the stature of invention beyond the other person's inventive concepts.

One cannot use the patent system to give a person working under him, with him, or over him a "pat on the back" or credit for a job done when that person employs only the skill normally expected of one with the usual skill in the art. Similarly, one cannot use the patent system to maintain a more congenial relationship with a vendor or vendee, a supervisor, or co-worker by handling the

invention as if such person were the inventor. The actual facts must speak for themselves and let the finding of inventorship fall where it must.

Correction of Errors in Inventors Named

As noted in the article on patents in the January-February-March 1958 issue of the General Motors Engineering Journal[1], when inventions are made by two or more persons, they may obtain a patent for their joint invention, but none of them can obtain a valid patent for the invention as the sole inventor. Certain new statutory provisions with respect to cases of joint inventorship were enacted by Congress in The Patent Act of 1952. Under these provisions, in some cases, a patent application erroneously filed because of errors in determining inventorship, or an issued patent maturing from such an application, can now be corrected.

If a person is joined in a patent application as joint inventor, or a joint inventor is not included in the application, *through error arising without any deceptive intent*, the application or patent may be appropriately amended to remove the name of the person not a joint inventor or to include the omitted joint inventor. More than one erroneously joined person may be omitted, or more than one erroneously omitted inventor may be added. The Commissioner of Patents has established a requirement of a verified statement of the facts, together with a new oath and diligence, in order to make such corrections. Thus, an application filed by *A* and *B* as joint inventors can, under the appropriate circumstances, be amended to become an application of *A* and *C*, or *A*, *B* and *C* as joint inventors, or an application of *A* alone. Likewise, an application filed by *A* as a sole inventor can be changed to a joint application of *A* and *B*. However, one cannot modify an application filed by *A* as the sole inventor, or by *A* and *B* as joint inventors, so that it becomes an application of *C* as the sole inventor or *C* and *D* as joint inventors.

An essential element of this procedure is that there must be no deceptive intent either when originally filing the application or when applying for correction. Questionable circumstances surrounding an invention which could possibly lead to a conclusion that there has been deceptive intent should be scrupulously avoided. Another essential element is that corrections must be requested promptly after the error is discovered.

Every effort should be made to determine inventorship correctly before the application is filed. Unusual occurrences, such as the filing of a vast majority of applications naming the supervisor of a group of engineers as inventor of ideas originating within the group, should be carefully investigated. Similarly, cases in which it is claimed that there is joint inventorship among more than two or three persons should be carefully investigated. In both such extremes it is well to remember that one is doing a disservice to himself, his employer, fellow employes, and to the public if he executes the oath required of all patent applicants, but is not the true inventor.

Filing Applications without Inventor's Cooperation

The present statute also takes care of certain situations, not previously correctible, in which a joint inventor refuses to join in an application for patent, or cannot be found or reached after diligent effort. Under these circumstances the application may be made by the other joint inventor on behalf of himself and the omitted inventor. After due proof of the pertinent facts and legal notice to the omitted inventor, the Commissioner of Patents is authorized to issue the patent to the inventor who made the application, subject to the same rights which the omitted inventor would have had if he had been joined.

Another relinquishment of the prior rigid requirement for the filing of the application by the inventor permits the filing of an application by a person other than the inventor in certain special situations. If an inventor cannot be found or reached after diligent effort has been made, or refuses to execute an application for patent, a person or officer of a corporation to whom the inventor has assigned the invention, or who can show sufficient proprietary interest in the invention, may apply for the patent on behalf of and as the agent for the inventor. These facts must be proven and it must be shown that such action is necessary to preserve the rights of the parties or to prevent irreparable damage. This is permitted to prevent the loss of rights and not to settle disputes. If a patent is granted on such an application, it is granted to the inventor and not to the person who made the application. If the applicant claims title, he must still take the necessary steps to obtain an assignment of the patent if he wishes to have the patent issued to him as assignee, or if he wishes to own the patent.

Applications for patents on inventions of deceased and legally incapacitated inventors may be made by their legal representatives.

A later filed application which is carved out of a pending application may be known as a *divisional application*. Under certain circumstances, a divisional application may be filed without the signature of the inventor. Application for reissue of a patent may be made by the assignee.

Bibliography

1. Barnard, Richard P., "A Dilemma of Group Research—Determination of 'Inventorship,'" *General Motors Engineering Journal*, Vol. 5, No. 1, January-February-March 1958, p. 40.

Notes About Inventions and Inventors

Contributed by Patent Section Detroit Office

The following is a general listing of patents granted in the names of General Motors employes during the period October 1, 1960, through December 31, 1960.

AC Spark Plug Division
Flint, Michigan

• **Joseph Zubaty,** (*M.S.M.E., University of Prague, 1918*) staff engineer—special assignment, inventor in patent 2,955,475 for a variable pressure fluid pump.

• **Frank B. Paul,** (*General Motors Institute, 1929*) senior designer, inventor in patent 2,956,462 for a spark plug socket wrench.

• **Joseph B. Kripke**, (*B.S.M.E., Massachusetts Institute of Technology, 1940*) staff engineer, Milwaukee Plant, inventor in patent 2,957,360 for a speed selector device.

• **Arnold B. Raninen**, (*B.S.M.E., Michigan College of Mining and Technology, 1940 and M.M.E., University of Delaware, 1951*) director, Mace engineering, Milwaukee Plant, inventor in patent 2,957,384 for an optical sighting device.

• **Virgil L. Helgeson**, (*B.E.E., University of Minnesota, 1948 and M.S.E.E., Washington University, 1950*) senior engineer, Milwaukee Plant, inventor in patent 2,957,535 for a vehicle steering control system with lateral acceleration computer.

• **William F. Thornburgh**, (*B.S.M.E., Michigan State University, 1951*) project engineer, inventor in patent 2,959,248 for an air cleaning and induction system inlet means.

• **Henry H. Harada**, senior designer, inventor in patent 2,959,914 for a drum type electric clock mechanism.

• **Robert W. Smith**, (*Ph.D., Physics, University of Michigan, 1933*) staff scientist; **Karl Schwartzwalder**, (*B.Cer.E., 1930, and M.S., 1931, The Ohio State University*) director of research, and William E. Counts, no longer with GM, inventors in patent 2,962,452 for ceramic semiconductor compositions.

• **Alfred Candelise**, (*Ph.D. in electrical-mechanical engineering, University of Naples, 1923*) staff engineer, inventor in patent 2,962,543 for a spark plug seal.

• **Raymond E. Summerer**, (*B.S.E.E., University of Colorado, 1951*) senior project engineer, inventor in patent 2,962,703 for a triple condition tell tale system.

• **Ronald J. Sargent**, (*B.M.E., General Motors Institute, 1956*) sales correspondent, inventor in patent 2,964,060 for a selector control.

• **Harry C. Zeisloft**, (*B.S.M.E., University of Iowa, 1941*) staff engineer, inventor in patent 2,965,076 for a servo mechanism.

Allison Division
Indianapolis, Indiana

• **Frederick W. Hoelje**, (*B.S.M.E., Bradley University, 1940*) senior design engineer, inventor in patent 2,929,663 for a bearing oil scavenger.

• **John R. Hayes**, (*General Motors Institute, 1944*) senior project engineer, inventor in patent 2,930,580 for a two-piece turbine bucket and in patent 2,942,311 for a turbine blade lock.

• **Edwin C. Schunke**, (*B.S.E.E., North Dakota State University, 1936*) head, Temperature, Measurement, and Ignition Department, inventor in patent 2,930,827 for a thermocouple.

• **Gerald E. Hook**, (*B.A., 1947, Elon College; B.S. Gen. Engr., 1949, and M.S. in Mathematics, 1951, North Carolina State College*) senior project engineer, and Elton K. Morice, not with GM, inventors in patent 2,926,529 for a propeller blade dynamic balance testing machine.

• **Mark E. Fisher**, (*B.S.M.E., Purdue University, 1947*) senior project engineer; **Robert H. Schaefer**, (*M.E., University of Munich, 1926*) manager, Transmission Engineering Department; and **Robert M. Tuck**, (*B.M.E., General Motors Institute, 1947*) chief development engineer, Transmission Engineering Department, inventors in patent 2,933,172 for a transmission clutch control system.

• **Chester E. Hockert**, (*B.S.M.E., Armour Institute, 1937, and M.S.M.E., Cornell University, 1941*) chief engineer, Turbo-Jet Engineering Department, and **Leslie R. Smith**, (*International Correspondence School*) chief draftsman, Turbo-Jet Engineering Department, inventors in patent 2,935,296 for a blade retaining means.

• **William F. Egbert**, (*B.S.Aero.E., Tri-State College, 1941*) group head, Exhaust and Fabricated Sections, and **Victor W. Peterson**, (*B.S.M.E., Rose Polytechnic Institute, 1939*) engineer, inventors in patent 2,963,857 for a turbojet engine.

• **Paul Bancel**, (*B.A., Wesleyan University, 1937*) engine project engineer, and **Victor W. Peterson***, inventors in patent 2,936,-655 for a self aligning planetary gearing.

• **Robert M. Tuck***, and Fred Snoy, no longer with GM, inventors in patent 2,936,865 for a transmission and control system.

• **John M. Wetzler**, (*B.S.M.E., The Johns Hopkins University, 1939*) section head, Turbo-Jet Design Group, inventor in patent 2,938,333 for a combustion chamber liner construction.

• **Albert J. Sobey, Jr.**, (*B.M.E., General Motors Institute, 1945*) section chief, rocket propulsion, and Phillip J. Brundage, no longer with GM, inventors in patent 2,957,424 for a centrifugal pump.

• **Robert G. Larkin**, (*B.S.M.E., Rose Polytechnic Institute, 1944*) senior project engineer; **Charles J. McDowall**, (*B.S. M.E., University of Florida, 1927*) technical assistant to the director of engineering; and Marion W. Dininger, no longer with GM, inventors in patent 2,959,228 for a torque responsive propeller control.

• **Floyd J. Boyer**, (*B.S.M.E., Georgia Institute of Technology, 1940*) group project engineer; **William R. Harding**, (*General Motors Institute, 1946*) supervisor, Production Control Systems; **Edmund M. Irwin**, (*B.S. in physics, Central Michigan College, 1938*) head, control electronics; **Charles J. McDowall***; and **Robert J. Wente**, (*B.S.M.E., Purdue University, 1941*) head, advanced design power turbine controls, inventors in patent 2,938,340 for a temperature datum gas turbine control.

• **Raymond Reger**, project engineer, inventor in patent 2,938,541 for a valve.

• **Albert J. Sobey, Jr.***, inventor in patent 2,941,354 for a variable jet nozzle control with auxiliary control of guide vanes and compressor bleed.

• **Howard W. Christenson**, (*B.S., Oregon State College, 1938*) head, Research Department, inventor in patents 2,930,257 and 2,965,202, both for a transmission.

• **Roy H. Brandes**, senior experimental engineer, and **Richard A. Hirsch**, (*University of Dayton and Sinclair College*) project engineer, inventors in patent 2,955,663 for a propeller control system.

***Inventors' names marked with an asterisk have biographical listings noted previously in this issue's Notes About Inventions and Inventors.**

• **Howard W. Christenson***, and **Ulysses A. Breting**, senior reliability engineer, Transmission Operations, inventors in patent 2,941,639 for a clutch and brake for steering vehicle.

• **Kenneth O. Johnson**, (*B.S.Aero.E., Purdue University, 1950*) senior project engineer, inventor in patent 2,941,848 for a spring load bearing support.

• **George B. Meginnis**, (*B.S.M.E., Purdue University, 1940*) senior project engineer, inventor in patent 2,942,311 for a method and apparatus for stripping casting.

• **Elroy P. Neate**, (*B.S.M.E., Purdue University, 1942*) overhaul supervisor, inventor in patent 2,942,844 for a turbine nozzle.

• **James N. Tootle**, (*B.S.A.E., University of Michigan, 1943*) supervisor—actuator design, inventor in patent 2,932,206 for a twin rotary actuator.

• **Roy H. Brandes***, and **Oren F. Flaugh**, (*B.S.M.E., Tri-State College, 1931*) designer, inventors in patent 2,940,426 for a volume compensating means for a servo system.

• **Arthur W. Gaubatz**, (*B.S., University of Wisconsin, 1920*) senior project engineer, inventor in patents 2,955,536 and 2,955,-537, both for a fuel pump; 2,955,542 for a vane pump; 2,955,609 for a dual pump fuel system; 2,960,327 for a speed switch actuating mechanism; 2,934,296 for a supported body with vibration absorbing resilient cantilever mount; and 2,934,367 for an accessory mount.

Buick Motor Division
Flint, Michigan

• **Forest R. McFarland**, (*B.S.M.E., Michigan State University, 1921*) executive assistant chief engineer, inventor in patent 2,963,289 for a fluid supply system.

• **Stanley L. Buckay**, (*B.S.M.E., Lawrence Institute of Technology, 1942*) staff engineer, and **Oliver K. Kelley**, (*B.S., Chicago Technical College, 1925 and Massachusetts Institute of Technology*) now technical assistant to the general manager, Defense Systems Division, inventors in patent 2,963,923 for an accessory drive.

• **Harold Fisher**, (*B.M.E., General Motors Institute, 1947*) section engineer, inventor in patent 2,964,963 for a transmission and control therefor.

• **Richard W. Craig**, (*B.M.E., General Motors Institute, 1949*) senior project engineer, inventor in patent 2,964,964 for a control mechanism.

• **Eric R. Dietrich**, senior project engineer, inventor in patent 2,965,389 for a fluid suspension control mechanism.

• **Harold H. Williams**, (*Michigan State University*) senior project engineer, and **William D. Pittsley**, (*B.S.M.E., University of Michigan, 1945*) senior project engineer, air cleaner engineering, AC Spark Plug Division, inventors in patent 2,965,404 for a slotted bumper exhaust device.

Business Research Staff
Detroit, Michigan

• **Andrew T. Court**, (*A.B., Vanderbilt University, 1924*) economist, inventor in patent 2,963,936 for a rear vision means having prismatic window.

Cadillac Motor Car Division
Detroit, Michigan

• **Casimer J. Cislo**, (*Wayne State University and University of Michigan*) senior project engineer, inventor in patents 2,954,850 and 2,962,297 for an actuating mechanism for disc brake and pneumatic suspension lift control, respectively.

• **Bruce M. Edsall**, (*B.S.M.E., Wayne State University, 1942*) staff engineer, inventor in patent 2,956,448 for torque converters combined with planetary gearing.

• **Harry M. Purdy**, senior project engineer, inventor in patent 2,963,300 for a vehicle frame.

Chevrolet Motor Division
Detroit, Michigan

• **Robert R. Parks**, (*B.M.E., General Motors Institute, 1957*) senior project engineer, and **Wayne A. Weaver**, (*Michigan State University-Oakland*) senior layout draftsman, inventors in patent 2,946,869 for a safety steering wheel and horn blowing means.

• **Zora Arkus-Duntov**, (*M.E., Institute of Charlottenburg, Berlin, Germany, 1934*) staff engineer, inventor in patents 2,947,293 and 2,949,102 for a fuel injection manifold and cold enrichment device, respectively.

• **Richard E. Denzer**, (*B.M.E., and M.S., The Ohio State University, 1951*) design engineer, and **Phillip C. Bowser**, (*B.M.E., The Ohio State University*) director, research and development, Buick Motor Division, inventors in patent 2,950,104 for an air spring.

• **Adelbert E. Kolbe**, (*University of Michigan*) assistant staff engineer, inventor in patent 2,951,391 for a distributor drive and thrust bearing therefor; patent 2,953,126 for an engine coolant distribution; and patents 2,963,012 and 2,957,-460, both for an internal combustion engine.

• **Gibson O. Hufstader**, (*General Motors Institute, 1955*) project engineer, inventor in patents 2,953,001 and 2,953,910, both for a universal joint expansion ball.

• **Eugene B. Etchells**, (*B.S.E.E., University of Michigan, 1932*) assistant staff engineer, inventor in patent 2,953,336 for an engine mounting system.

• **Frank J. Winchell**, (*Purdue University*) director, research and development, inventor in patents 2,955,482 and 2,965,205, both for a transmission.

• **Bruno J. Olender**, (*Detroit City College*) staff engineer, inventor in patent 2,955,-863 for a tail gate latch for pick-up truck box.

• **Richard G. White**, (*B.M.E., General Motors Institute, 1949*) project engineer, inventor in patent 2,964,351 for an adjustable license plate assembly.

• **George J. Englehard**, (*B.I.E., General Motors Institute, 1952*) assistant staff engineer, inventor in patent 2,965,414 for a vibration dampening vehicle frame construction.

• **George H. Primeau,** (*Northern State Teachers College*) design engineer, inventor in patent 2,959,067 for a transmission control lever bearing.

• **Maurice S. Rosenberger,** (*Nebraska Wesleyan University*) assistant chief engineer in charge of engine and passenger car chassis design, inventor in patent 2,962,011 for a mechanical valve lash adjuster.

• **Lawrence N. Reed,** (*University of Detroit*) senior specialized clerk, inventor in patent 2,962,073 for a resilient mounting for a valve stem assembly or the like.

• **Joseph B. Depman,** (*B.S.M.E., Villanova University, 1950*) resident product engineer, inventor in patent 2,962,110 for a surge muffler for air storage tank.

• **William S. Wolfram,** (*B.S.M.E., University of Michigan, 1933*) assistant staff engineer, inventor in patent 2,962,910 for an accessory drive.

• **Earl W. Rohrbacher,** (*B.S.M.E., University of Utah, 1926*) design engineer, inventor in patent 2,963,111 for an air cleaner silencer assembly.

• **Robert W. Graham,** (*B.S.M.E., University of Michigan, 1949*) senior project engineer, inventor in patent 2,964,120 for a high pressure storage tank inlet fitting and muffler assembly.

Delco Appliance Division
Rochester, New York

• **Francis M. Ryck,** (*B.S., University of Rochester, 1950*) assistant supervisor, windshield wiper applications, inventor in patents 2,964,776 and 2,958,892 for a windshield wiper blade and a wiper arm connector attachment means, respectively.

• **William E. Fritz,** (*Technical College of Thurengia, Hildburghausen, Germany, 1928*) senior development engineer, inventor in patent 2,964,933 for a program timer.

• **Eugene R. Ziegler,** (*University of Rochester*) special development engineer, inventor in patents 2,965,913 and 2,959,-803, both for a windshield cleaning system and also patent 2,956,299 for a connector for windshield wiper.

• **Frederick Druseikis,** senior designer, inventor in patent 2,956,622 for a gas burner.

• **Loren R. Gute,** (*B.S.E.E., Michigan State University, 1940*) supervisor of defense products, inventor in patent 2,957,056 for switch assemblies.

• **John G. Hart,** (*B.S.M.E., University of Rochester, 1949*) presently on educational leave of absence, and **Kenneth A. Kosbab,** (*B.S.M.E., University of Rochester, 1951*) project engineer, inventors in patent 2,957,193 for a control mechanism for windshield wipers and washers.

• **John G. Hart*** and **Eugene R. Ziegler***, inventors in patent 2,958,891 for a windshield washer.

• **Loren R. Gute*; Walter D. Harrison,** project engineer; and Harry W. Schmitz, no longer with GM, inventors in patent 2,959,968 for a windshield wiper actuating mechanism.

Delco Moraine Division
Dayton, Ohio

• **Ronald L. Shellhause,** (*B.S.M.E., Tri-State College, 1953*) project engineer, inventor in patent 2,955,887 for a piston seal structure.

• **Elton S. Moyer,** (*General Motors Institute, 1933*) chief engineer, inventor in patent 2,958,198 for a hydraulic actuating system.

• **Frederick W. Sampson,** (*M.E., Cornell University, 1924*) section engineer on special assignment, inventor in patent 2,966,239 for a brake structure.

Delco Products Division
Dayton, Ohio

• **James E. Whelan,** (*B.M.E., University of Illinois, 1951*) project engineer, inventor in patents 2,954,987 and 2,965,391, both for a two-position air suspension control valve.

• **John F. Pribonic,** (*B.S.M.E., Princeton University, 1947*) staff engineer, inventor in patents 2,956,816 for a two-position air suspension control valve; 2,957,704 for a fluid delayed valve control for suspension system; and 2,965,200 for a damper for valve control.

• **Harold E. Schultze,** project engineer and **Lawrence L. Evert,** (*B.S.M.E., Notre Dame University, 1948*) senior product engineer, Euclid Division, inventors in patent 2,957,456 for a grease device.

• **Albert W. Kolbe, Jr.,** (*B.M.E., General Motors Institute, 1956*) senior process engineer, inventor in patent 2,958,835 for a pickup instrument for measuring device.

• **George W. Jackson,** (*B.S.M.E., Purdue University, 1937*) assistant chief engineer and **John T. Hoban,** (*B.M.E., University of Dayton, 1943*) assistant staff engineer, Cadillac Motor Car Division, inventors in patent 2,962,296 for a fluid suspension and control system therefor.

Delco Radio Division
Kokomo, Indiana

• **Kenneth S. Vogt,** (*The Ohio State University*) senior project engineer, inventor in patent 2,958,032 for a transistor inverter and half-wave rectifier circuit.

Delco-Remy Division
Anderson, Indiana

• **Brooks H. Short,** (*B.S.E.E., 1931 and M.S.E.E., 1934, Purdue University*) director of Advanced Engineering, inventor in patent 2,955,248 for an ignition system.

• **John W. Dyer,** (*B.S.E.E., Tri-State College, 1941, and Purdue University*) research engineer, and **Richard L. Sprague,** (*B.S.E.E., Purdue University, 1953*) senior research engineer, inventors in patent 2,959,238 for a coincidental door locking apparatus for an automobile.

• **Julius H. Bolles,** (*B.S. in physics, Ohio Wesleyan University, 1920*) director of product reliability; Robert O. Hartman, no longer with GM; and **Brooks H. Short***, inventors in patent 2,964,774 for a windshield wiper mechanism.

• **James C. Norris**, (*Purdue University*) head, Ignition Section, and **John S. Oliver**, layout man, inventors in patent 2,965,726 for a distributor.

• **Julius H. Bolles***; **Lyman A. Rice**, (*B.S.E.E., University of Utah, 1935, and M.S.E., University of Michigan, 1936*) staff engineer, and Paul L. Schneider, deceased, inventors in patent 2,965,765 for a battery charging circuit.

Detroit Diesel Engine Division
Detroit, Michigan

• **Mark L. Beardslee**, (*B.S.M.E., Michigan State University, 1929*) senior research engineer; **James L. Stone**, senior project engineer; and Walter N. Frank, retired, inventors in patent 2,955,327 for a method for sealing electrical switches or the like.

• **Albin Chaplin**, (*B.S.M.E., Detroit Institute of Technology, 1948*) senior designer and **Edward A. Chapin**, (*B.S.Aero.E., 1934, and M.S.Aero.E., 1935, University of Michigan*) senior project engineer, inventors in patent 2,956,642 for a camshaft and bearing lubricating means.

• **John H. Smith**, (*B.S.M.E., University of Michigan, 1949*) application engineer, inventor in patent 2,960,082 for an engine starting and protective shutdown system.

Detroit Transmission Division
Ypsilanti, Michigan

• **Walter B. Herndon**, (*B.S.E., State College of Washington, 1928, and M.S.E., University of Michigan, 1930*) director of engineering and sales, and Victor C. Moore, no longer with GM, inventors in patent 2,957,373 for a dual coupling shift transmission.

• **Kenneth E. Snyder**, senior project engineer, inventor in patents 2,957,735 for a seal structure; 2,965,120 for a pressure control device; and 2,965,207 for a piston travel controlling mechanism.

• **Darrel R. Sand**, (*B.M.E., General Motors Institute, 1949*) assistant staff engineer, inventor in patent 2,964,135 for a brake mechanism.

• **Jack W. Qualman**, (*General Motors Institute, 1937*) assistant chief engineer, and Ralph F. Beck, retired, inventors in patent 2,964,959 for an accessory drive transmission.

• **John Chimko**, (*M.E. Diploma, International Correspondence School, 1956*) senior project engineer, and **Louis M. Fiteny**, (*B.S.E., University of Michigan, 1948*) assistant staff engineer, inventors in patent 2,965,189 for a vehicle speed governing device for use with automatic transmissions.

Diesel Equipment Division
Grand Rapids, Michigan

• **Philip W. Howson**, designer, inventor in patent 2,962,012 for a horizontally operable hydraulic valve lifter.

• **Albert F. Davis**, (*B.S.M.E., Notre Dame University, 1928*) general manager, and **Charles H. Palmer**, (*B.S.Chem.E., Michigan State University, 1935*) chief metallurgist, inventors in patent 2,963,011 for a valve lifter.

• **William G. Jacobitz**, (*B.S.M.E., Notre Dame University, 1953*) process engineer, and **Elias W. Scheibe**, (*B.S.M.E., University of Michigan, 1940*) senior project engineer, inventors in patent 2,963,282 for a fuel nozzle.

Electro-Motive Division
La Grange, Illinois

• **Arthur H. Juhlin**, senior project engineer, inventor in patent 2,955,882 for a railway vehicle brake system and valving therefor.

• **Albert N. Addie**, (*B.S.M.E., Illinois Institute of Technology, 1944, and M.S.M.E., Case Institute of Technology, 1947*) chief research engineer, and **Clarence H. Patrie**, (*B.S.M.E., University of Dayton, 1936*) senior designer, inventors in patent 2,960,354 for a pressure compensated flexible pipe.

• **Lauren L. Johnson**, (*B.S.E.E., University of Nebraska, 1938*) electrical control development engineer, inventor in patent 2,962,644 for a generator motor power train and control.

GM Engineering Staff
Warren, Michigan

• **Lothrop M. Forbush**, (*B.S., Harvard University, 1939, and Massachusetts Institute of Technology*) engineer in charge, Vehicle Development Group, and Ronald V. Hutchinson, retired, inventors in patent 2,955,426 for a power actuator for hydraulic brakes.

• **Oliver K. Kelley**, (*B.S., Chicago Technical College, 1925, and Massachusetts Institute of Technology*) now technical assistant to the general manager, Defense Systems Division; **Gilbert K. Hause**, engineer in charge, Transmission Development Group; and **Burnette Heck**, (*Wayne State University, 1935*) senior project engineer, inventors in patent 2,955,682 for a liquid cooled friction brake.

• **Harold L. Beck**, (*Port Huron Junior College*) layout man, and **Harold E. Boettger**, (*Wayne State University*) senior designer, inventors in patent 2,956,606 for a patch element for tubeless tires and the like.

• **Lothrop M. Forbush***; **Victor A. Rusnack**, (*B.M.E., University of Detroit, 1940*) now area engineering supervisor, New Departure Division; **Gerald Stofflet**, (*B.S.M.E., Detroit Institute of Technology, 1950*) senior layout man; and **John S. Wroby**, (*B.M.E., University of Detroit, 1942*) design engineer, inventors in patent 2,956,632 for a wheel hub assembly.

• **Von D. Polhemus**, (*B.S.M.E., University of Cincinnati, 1933*) engineer in charge, Structure and Suspension Development Group, inventor in patent 2,956,797 for a dual volume variable rate air spring.

• **Oliver K. Kelley***, and **Gilbert K. Hause***, inventors in patent 2,957,340 for a multi-phase torque converter.

• **Gilbert K. Hause***, and **Harold Fischer**, (*B.M.E., General Motors Institute, 1947*) section engineer, Buick Motor Division, inventors in patent 2,957,557 for a transmission.

• **John S. Wroby***, inventor in patent 2,962,279 for a resilient mounting for a vehicle suspension system.

• **Johannes Rosenkrands**, (*M.S.M.E., Royal Technical University of Denmark,*

1946) assistant engineer in charge, Structure and Suspension Development Group, inventor in patent 2,963,301 for a semi-independent vehicle wheel suspension.

Euclid Division
Cleveland, Ohio

• **John P. Carroll,** (*Bradley University*) chief product engineer; **Janis Mazzarins,** (*Technical University, Aachen, West Germany*) senior designer; **Edwin J. Selyem,** (*Fenn College and Case Institute of Technology*) now senior designer, Cadillac Motor Car Division; and **Russell C. Williams,** (*B.S.M.E., University of Illinois, 1932*) manager, research and test, inventors in patent 2,955,845 for a twin engine crawler tractor.

Fisher Body Division
Warren, Michigan

• **William G. Blenman,** (*B.M.E., University of Detroit, 1952*) senior production engineer, and **William A. Brady, Jr.,** (*Lawrence Institute of Technology*) senior process engineer, inventors in patent 2,955,568 for paint spraying apparatus.

• **David D. Campbell,** (*B.M.E., General Motors Institute, 1953*) assistant engineer in charge, Experimental and Development—Mechanical Design Group, and **Louis P. Garvey,** (*B.M.E., University of Detroit, 1940*) assistant engineer in charge, Product Engineering Activity, Design and Drafting Department, inventors in patent 2,955,817 for a window regulator mechanism.

• **James D. Leslie,** (*B.M.E., University of Detroit, 1939*) engineer in charge, Mechanical Department, and **Gerhard Rehkugler,** designer, inventors in patent 2,955,865 for an automobile door latch.

• **George Dominick,** project engineer, and **John E. Tessmar,** senior project engineer, inventors in patent 2,955,875 for a fabric attachment.

• **Harry C. Malpass,** (*General Motors Institute*) engineer in charge, Production Engineering Activity, inventor in patent 2,956,549 for a dual piston cylinder.

• **George D. Legge,** (*Western Technical Institute, Toronto, Canada*) senior project engineer, inventor in patent 2,957,073 for a combination dome and reading lamp assembly.

• **Engelbert A. Meyer,** senior project engineer, inventor in patent 2,959,259 for a fastener device.

• **Alfons A. Limberg,** (*Automotive Engineering School, Berlin, Germany*) now chief engineer, body and truck exterior, GM Styling Staff, inventor in patent 2,959,808 for a reciprocating door hinge.

• **Louis P. Garvey*,** inventor in patent 2,960,362 for a vehicle door latch and latch control.

• **Clyde H. Schamel,** (*B.S.E.E., University of Notre Dame, 1927*) engineer in charge, Central Experimental and Developmental Department, and **Claud S. Semar,** (*Detroit City College, University of Michigan, and Wayne State University*) senior project engineer, inventors in patent 2,963,075 for a seat adjuster.

• **Edward MacCallum,** (*Wayne State University*) group leader, inventor in patent 2,963,133 for a clip.

• **Edward G. Doyle,** (*B.S.M.E., Detroit Institute of Technology, 1946*) assistant engineer in charge, Design and Drafting, and **Herman A. Klix,** (*Chrysler Institute*) group leader, inventors in patent 2,964,341 for a pivotal joint.

• **Napolean P. Boretti,** (*B.E.E., University of Detroit, 1935*) assistant engineer in charge, Process Development Department, inventor in patent 2,964,611 for pressure welding apparatus.

• **Roy T. Collins,** (*General Motors Institute, 1939*) senior project engineer, and **Edward K. Tazzia,** (*B.S., Wayne State University, 1950*) packaging engineer, inventors in patent 2,965,346 for a freight car wall bracket.

These patent listings are informative only and are not intended to define the coverage which is determined by the claims of each one.

Frigidaire Division
Dayton, Ohio

• **Millard E. Fry,** (*B.S.M.E., University of Pittsburgh, 1931*) senior project engineer, inventor in patent 2,955,189 for a domestic appliance.

• **LeRoy McDonald,** (*B.S.M.E., University of Texas, 1954*) project engineer, inventor in patent 2,955,891 for a domestic appliance.

• **Kenneth O. Sisson,** (*B.S.M.E., South Dakota State College, 1936*) senior project engineer, inventor in patent 2,956,426 for a domestic appliance.

• **John R. Johnston,** (*B.S.E.E., Purdue University, 1934*) senior engineer, and **Daniel J. Barbulesco,** (*B.S.M.E., Stanford University, 1949*) project engineer, inventors in patent 2,956,684 for a domestic appliance.

• **Everett C. Armentrout,** (*B.S. in physics, University of Dayton, 1957*) project engineer, inventor in patent 2,957,320 for a refrigerating apparatus with magnetic door seal actuated switch.

• **John C. Rill, Jr.,** (*B.A., economics, Brown University, 1952*) project engineer, inventor in patent 2,958,210 for refrigerating apparatus.

• **Leonard J. Mann,** (*M.E., University of Cincinnati, 1940*) senior project engineer, inventor in patent 2,958,934 for a method of making refrigerating apparatus.

• **Francis S. Longnecker,** (*General Motors Institute*) project engineer, inventor in patent 2,958,954 for a laundry drier with sprinkling device.

• **Leonard J. Mann*,** and **Edward C. Simmons,** (*University of Dayton*) senior engineer, inventors in patent 2,949,936 for refrigerating apparatus.

• **John J. O'Connell,** (*B.S., physics, Xavier University, 1950*) senior project engineer, inventor in patent 2,960,849 for refrigerating apparatus.

• **George B. Long,** (*B.S.E.E., Purdue University, 1937*) supervisor of major product line, inventor in patent 2,961,520 for a domestic appliance.

• **John C. Rill, Jr.***, and **Keith K. Kesling**, (*University of Dayton and Dayton Art Institute*) project and design engineer, inventors in patent 2,963,183 for a refrigerator cabinet.

• **James A. Canter**, (*B.M.E., The Ohio State University, 1936*) senior project engineer, inventor in patent 2,964,922 for refrigerating apparatus.

• **George C. Pearce**, (*B.S.M.E., Stanford University, 1924*) section head, Non-Refrigerated Appliances Engineering Department and **Lester M. Miller**, (*Dayton Art Institute, Art Academy of Cincinnati, and the Central Academy of Commercial Art*) junior engineer, inventors in patent 2,965,095 for a ventilated superposed oven.

• **John Weibel, Jr.**, (*B.S.M.E., Louisiana State University, 1948, and M.S.M.E., Purdue University, 1950*) senior project engineer, inventor in patent 2,965,289 for a motor-compressor support.

• **James W. Jacobs**, (*B.S.M.E., University of Dayton, 1954*) manager, research and future products engineering, inventor in patent 2,965,317 for a domestic appliance.

• **James W. Jacobs***, and Francis H. McCormick, retired, inventors in patent 2,965,428 for a domestic appliance.

GMC Truck and Coach Division
Pontiac, Michigan

• **Hans O. Schjolin**, (*B.S. degree, Karlstad College, Sweden, 1920, and Polytechnical Institute, Mittweida, Germany, 1923*) staff engineer, and **Donald K. Isbell**, senior engineer, inventor in patent 2,955,678 for a vehicle disc brake.

• **John G. Locklin**, (*B.S. Aero.E., Detroit Institute of Technology, 1945*) engineer, truck chassis section, inventor in patent 2,955,814 for an air spring assembly.

Guide Lamp Division
Anderson, Indiana

• **Lloyd T. Fuqua**, (*DePauw University and the American School of Chicago*) senior designer, inventor in patent 2,964,350 for adjustable license plate assembly.

• **George W. Onksen**, (*B.I.E., General Motors Institute, 1956 and Purdue University*) staff engineer—reliability; **Harold E. Todd**, (*B.S.E.E.*, Purdue University, 1940) sales engineer; and **Myrneth L. Woodward**, project engineer, inventors in patent 2,965,813 for an automatic headlight dimmer system.

Hyatt Bearings Division
Harrison, New Jersey

• **Ralph A. Altson**, (*teacher's diploma, Regent Technical Institute, London, England, 1930*) chief design engineer, inventor in patent 2,958,051 for a dimensional tolerance gauge.

• **Theodore Jagen**, (*M.E., Stevens Institute of Technology, 1935*) director, production engineering, inventor in patent 2,961,744 for a method of making bearing rings.

Inland Manufacturing Division
Dayton, Ohio

• **Max P. Baker**, (*A.B., Miami University, 1922*) project engineer, inventor in patent 2,954,992 for a ball and socket joint assembly and method of making same.

• **Paul E. Clingman**, *General Motors Institute, 1935*) supervisor, quality and control distribution, inventor in patent 2,956,705 for cabinet framework.

• **James R. Wall**, (*B.Ch.E., University of Dayton, 1937 and M.Ch.E., Cornell University, 1939*) supervisor, Advanced Development Laboratory, and **Murray S. Millhouse**, (*General Motors Institute, 1941*) project engineer, inventors in patent 2,958,516 for apparatus for making synthetic resins.

Oldsmobile Division
Lansing, Michigan

• **Kenneth E. Faiver**, (*B.S.E.E., Notre Dame University, 1924 and Ph.D., Rensselaer Polytechnic Institute, 1927*) senior project engineer, and **Ralph W. Perkins**, (*B.S. M.E., Purdue University, 1937*) chassis engineer, inventors in patent 2,955,841 for a wheel hop damper.

• **Albert D. Baker**, (*B.S.M.E., Purdue University, 1926*) heating, cooling, and ventilating engineer, inventor in patent 2,963,954 for automotive heating, ventilating and defrosting systems.

• **Robert Seyfarth**, (*B.S.M.E., Princeton University, 1936*) senior engineer, engine group, inventor in patent 2,965,871 for a spark plug lead terminal connection.

GM Overseas Operations Division
New York, New York

• **Eric W. A. Smith**, (*Luton College of Technology*) project engineer, and **Frank A. Rowley**, (*Luton College of Technology*) senior engineer, Vauxhall Motors, Limited, Luton, England, inventors in patent 2,949,523 for a dielectric heating apparatus.

• **Kenneth E. Buckman**, assistant chief engineer, No. 2 Plant, AC-Delco Division, Southampton, England, inventor in patent 2,945,559 for filters for fluids.

• **Albert A. Kenlock**, (*Regent Street Polytechnic*) assistant advance design engineer, and **John Alden**, (*School of Technology, Oxford*) chief product engineer, Vauxhall Motors, Limited, Luton, England, inventors in patent 2,959,230 for truck cabs.

Packard Electric Division
Warren, Ohio

• **Elbert L. Johnson**, (*General Motors Institute, 1936*) head, automotive application engineering, and **William J. Jensen**, Pontiac Motor Division, inventors in patent 2,962,689 for a battery terminal connector.

• **Richard J. Lander**, (*Youngstown University*) project engineer; **Robert H. Sims**, (*B.S.I.E., Illinois Institute of Technology, 1948*) Detroit district sales manager; and **Robert C. Woofter**, (*Fenn College*) chief, Wiring Assemblies Design and Development Section, inventors in patent 2,955,-178 for a fuse panel assembly.

Pontiac Motor Division
Pontiac, Michigan

• **Clayton B. Leach,** (*A.B. in mathematics and chemistry, Park College, 1934, and General Motors Institute*) chassis engineer, inventor in patent 2,955,675 for an engine lubricating system and patent 2,963,007 for an engine with reversible heads, couplings, and gaskets.

• **John Z. DeLorean,** (*B.S.I.E., Lawrence Institute of Technology, 1948; M.S.A.E., Chrysler Institute, 1952; M.B.A., University of Michigan, 1957; and Detroit College of Law*) assistant chief engineer in charge of advanced design and body, inventor in patent 2,960,178 for an engine exhaust system and patents 2,961,896 and 2,964,-975, both for a transmission.

GM Research Laboratories
Warren, Michigan

• **Robert F. Falberg,** (*B.S.M.E., Michigan College of Mining and Technology, 1958*) senior research engineer, inventor in patent 2,956,396 for a free piston engine starter.

• **Frederick W. Chapman,** (*B.S.E.E., University of Michigan, 1938*) senior research engineer, inventor in patent 2,957,081 for a radiator detector.

• **James G. Roberts,** (*B.S.Civil Engr., University of Illinois, 1949*) senior research engineer, inventor in patent 2,958,925 for a shot peen inspection technique.

• **John S. Collman,** (*B.S., Naval Architecture and Marine Engineering, University of Michigan, 1940*) assistant head, Engineering Development Department, and **James M. Ricketts,** (*General Motors Institute, 1944*) supervisor, engineering design, inventors in patent 2,960,306 for a turbine.

• **Joseph B. Bidwell,** (*B.S.M.E., Brown University, 1942*) head, Engineering Mechanics Department, inventor in patent 2,962,108 for a servo system with controls for vehicular power steering.

• **Edward F. Weller, Jr.,** (*B.S.E.E., University of Cincinnati, 1943*) assistant head, Physics Department, inventor in patent 2,962,892 for an automatic engine indicator plotter.

• **Darl F. Caris,** (*B.S.E.E., 1926, and professional degree of E.E., 1932, University of Michigan*) now engineer in charge, Power Development Group, GM Engineering Staff, inventor in patent 2,963,015 for an engine.

• **John M. Farrell,** project engineer, and Edward J. Martin, retired, inventors in patent 2,964,957 for an index mechanism.

• **Robert E. Osborne,** (*A.B. Chem. and Math., Kansas State Teachers College, 1941*) research engineer, inventor in patent 2,964,967 for a steering gear or the like.

• **Worth H. Percival,** (*B.S.M.E., Iowa State College, 1942, and M.S.M.E., Massachusetts Institute of Technology, 1947*) assistant head, Mechanical Development Department, inventor in patent 2,965,083 for accumulator supercharging.

• **Eugene A. Hanysz,** (*B.S.E.E., 1945, and M.S.E.E., 1948, University of Michigan*) supervisor, Physics Department, inventor in patent 2,965,188 for a vehicle control means.

• **Thomas J. Hughel,** (*B.S. Met. Eng., 1942, and Ph.D., 1951, Purdue University*) supervisor, Metallurgical Engineering Department, inventor in patent 2,966,033 for refrigerating apparatus.

Rochester Products Division
Rochester, New York

• **Lawrence C. Dermond,** (*Purdue University and Tri-State College*) staff engineer, inventor in patent 2,959,135 for a pump.

Saginaw Steering Gear Division
Saginaw, Michigan

• **Charles W. Spalding,** (*B.S.M.E., Michigan State University, 1940*) design engineer, and Walter H. West and Bryan E. Nixon, not with General Motors, inventors in patent 2,956,405 for a power system.

• **Philip B. Zeigler,** (*B.S.M.E., Purdue University, 1941*) chief engineer, Product Engineering Department and **George A. Edwards,** (*Fenn College*) design engineer, inventors in patent 2,956,586 for a hose arrangement.

• **Paul V. Wysong, Jr.,** (*University of Kansas*) chief applications engineer, inventor in patent 2,963,217 for a fluid compressor.

• **Donald P. Marquis,** (*B.S.Chem.E., 1934 and M.S., 1939, Wayne State University*) assistant chief engineer, inventor in patent 2,964,928 for a universal joint.

Ternstedt Division
Detroit, Michigan

• **Akira Tanaka,** (*B.S.M.E., Michigan State University, 1949*) design group leader, and **Nicholas Toruk,** (*B.S.M.E., University of Detroit, 1951*) senior project engineer, inventors in patent 2,959,208 for a seat adjusting mechansim.

• **Frank A. Croskey,** research engineer, and **Charles D. Tuttle,** (*Ph.D., Michigan State University, 1933*) senior experimental physicist, inventors in patents 2,959,353 and 2,960,273 for an electrostatic charger apparatus and electrostatic spray painting apparatus, respectively.

• **John P. Bogater,** (*B.S.M.E., Detroit Institute of Technology, 1937*) design group leader; **Bela Sandor,** (*B.S.M.E. and B.S. A.E., University of Engineering and Agricultural Academy, Hungary, 1932*) senior designer; and **Alfonsas Velavicius** (*University of Kansas and University of Detroit*) senior designer, inventors in patent 2,959,448 for a closure latch.

• **James H. Dozois,** (*B.S.M.E., University of Detroit, 1948*) engineering group supervisor, and **Frederick L. Schwartz,** (*Lawrence Institute of Technology*) senior body designer, Fisher Body Division, inventors in patent 2,962,751 for a door hinge assembly.

• **Thomas E. Lohr,** (*Tri-State College*) senior design engineer, and **Akira Tanaka***, inventors in patent 2,964,093 for a vehicle seat adjuster.

• **Barthold F. Meyer,** (*B.S.M.E., Pratt Institute, 1939 and Johns Hopkins University*) engineering group supervisor, inventor in patent 2,966,559 for a circuit controller.

• **Arthur W. Hollar, Jr.,** (*B.S.M.E., University of Michigan, 1941*) senior designer, inventor in patent 2,965,402 for remote locking button devices.

Technical Presentations by GM Engineers and Scientists

The technical presentation is another way in which information about current engineering and scientific developments in General Motors can be made available to the public. A listing of speaking appearances by General Motors engineers and scientists, such as that given below, usually includes the presentation of papers before professional societies, lecturing to college engineering classes or student societies, and speaking to civic or governmental organizations. Educators who wish assistance in obtaining the services of GM engineers and scientists to speak to student groups may write to the Educational Relations Section, Public Relations Staff, General Motors Corporation, General Motors Technical Center, Warren, Michigan.

The following GM personnel made recent technical presentations.

Automotive Engineering

J. D. Peebles, sales engineer, Allison Division, before a joint meeting of the American Society for Metals and the American Society of Mechanical Engineers, New York City, title: The Automotive Gas Turbine.

Mieczykslaw G. Bekker, group head—land operations, Defense Systems Division, before the A.S.M.E., New York City, title: The Evolution of Locomotion—A Conjecture into the Future of Vehicles.

Sheldon T. Smith, supervisor, engineering services, Delco Appliance Division, before the Fourth Order, Knights of Columbus, Rochester, New York, title: Automotive Power Assists.

William H. Jackson, superintendent, laboratories and shops, Harrison Radiator Division, before the 7th Annual Automotive Air Conditioning Forum, Dallas, Texas, title: Air Conditioning the Compact Car.

Richard F. Youngblood, senior project engineer, Oldsmobile Division, before the Michigan State University student section, A.S.M.E., title: Suspension Geometry.

Clayton B. Leach, chassis engineer, Pontiac Motor Division, before alumni of General Motors Institute, Indianapolis, Indiana, title: The Pontiac Tempest.

Frank Sciabica, senior project engineer, Rochester Products Division, before the Diesel technology class, Alfred University, Alfred, New York, title: Fuel Injection.

From AC Spark Plug Division: **Glen R. Fitzgerald**, director of engineering and equipment sales, before the California State Vehicle Pollution Control Board, Los Angeles, title: Positive Crankcase Ventilation; **George A. Brown**, field contact engineer, before the Motor Mercantile Company, Salt Lake City, title: Spark Plug Applications and Proper Maintenance; and **James H. DeVoe**, field engineer, before the Fleet Maintenance Association of Buffalo, New York, title: Oil Filtration.

From GMC Truck and Coach Division: **C. V. Crockett**, chief engineer, before the American Society of Tool and Manufacturing Engineers, Pontiac, Michigan, section, title: GMC V-6 and Twin Six Engines; **Frank C. Fleck**, truck chassis section engineer, before the Society for Nondestructive Testing, Philadelphia, title: The Use of Experimental Analysis in Truck Frame Design, and **K. L. Raymond**, laboratory engineer, before the Michigan Trucking Association, Detroit, title: Corrosion of Highway Vehicles.

From GM Engineering Staff: **Charles A. Chayne**, vice president in charge, before the Southern Research Institute Transportation Conference, Birmingham, Alabama, title: Technology and Tomorrow's Motor Vehicle, and before the Engineers Club of Dayton, Ohio, title: General Motors Smaller Cars, and **Craig Marks**, assistant engineer in charge, Power Development Group, before the A.S.M.E. Paducah chapter, Carbondale, Illinois, title: Piston Engines in the Space Age?

From the GM Research Laboratories: **C. A. Amann**, supervisor, Engineering Development Department, before the Detroit section, A.S.M.E., title: Ground Effect Devices; **W. D. McMaster**, assistant head, Chemistry Department, before the Chemical Specialities Manufacturers Association, Hollywood, Florida, title: Chemical Specialties and the Modern Automobile; and **A. J. Burkman**, supervisor, Engineering Mechanics Department, before the A.S.M.E.-A.S.L.E. Lubrication Conference, Boston, title: Moisture Sensitivity of Brake Lining Materials.

Before various S.A.E. student section meetings: **R. C. Balmer**, design engineer, GMC Truck and Coach Division, Pennsylvania State University, title: GMC V-6 and Twin Six Engines; **Thomas M. Fisher**, administrative engineer, GM Proving Grounds, Purdue University, title: Proving Ground Testing; **Joseph M. Sherwood**, general supervisor, engineering, Buick-Oldsmobile-Pontiac Assembly Division, Kansas City, Kansas plant, Missouri School of Mines and Metallurgy, title: Problems of Automobile Assembly; and **W. A. Turunen**, head, Engineering Development Department, GM Research Laboratories, Michigan College of Mines and Technology, title: Gas Turbine Progress.

Before various S.A.E. section meetings: **John Z. DeLorean**, assistant chief engineer, Pontiac Motor Division, before the Detroit section, title: The Pontiac Tempest; **Robert H. Knickerbocker**, senior project engineer, Pontiac Motor Division, before the Pittsburgh section, title: The Pontiac Tempest; **Roy M. Law**, experimental engineer, Detroit Diesel Engine Division, before the Cleveland section, title: New Techniques in Stress Analysis; **Karl Schuster**, project engineer, GMC Truck and Coach Division, before the Rockford-Beloit, Wisconsin, section, title: Air Conditioning of GMC Coaches; **Frank W. Sinks**, senior project engineer, Detroit Diesel Engine

Division, before the Chicago section, title: Turnpike Cruisers.

Bearings

J. G. Turnbull, sales engineer, Allison Division, before the Central Railroad Club, Buffalo, New York, title: Kar-Go Journal Bearings.

From New Departure Division: **C. R. Gillette,** manager, research chemistry, before the A.S.L.E., Newark, New Jersey, title: Lubrication of Anti-Friction Bearings; **C. T. Bragdon,** agricultural and textile engineer, before engineers of the Ford Tractor and Implement Company, Detroit, the Massey-Ferguson Company, Detroit, and the Allis Chalmers Company, Independence, Missouri, title: Progress in Seals for Implement Bearings; **C. G. Hayden,** general project engineer, before engineers of the Warner and Swasey Company, Cleveland, and the National Acme Company, Cleveland, title: Ball Bearing Selection and Application; and **R. B. Walker,** electric motor project engineer, before engineers of Robbins and Meyers Company, Springfield, Ohio, Remington Rand Company, Bridgeport, Connecticut, and Universal Electric Company, Owosso, Michigan, title: Effect of Ball Bearings on Electric Motor Sound.

Diesel Engines

From Detroit Diesel Engine Division: **C. Walter Frederick,** director of engineering, before the Fleet Maintenance Conference, New York City, title: Diesel Engine Developments; **Roger D. Wellington,** assistant staff engineer—director of test, before the Detroit section, S.A.E., title: Compression Temperatures in Diesel Engines Under Starting Conditions; and **William S. Kenyon, Jr.,** senior project engineer, before the S.A.E., subcommittee on valve seat inserts, Cleveland, title: Piston Rings for Transportation Diesels.

Electrical Engineering

Ralph K. Shewmon, assistant chief engineer, electrical products, Delco Products Division, before students of General Motors Institute, title: Technical Aspects of Electric Motor Application.

J. B. Bidwell, head, Engineering Mechanics Department, GM Research Laboratories, before the American Association of State Highway Officials, Detroit, title: Electronic Applications in Vehicle Design and Operation.

From Delco Radio Division: **A. L. Anderson,** project engineer, before the central Indiana section, Institute of Radio Engineers, Indianapolis, title: Modular Building Blocks in Digital Systems; **J. O. Beasley,** field service engineer, before the National Alliance of Television and Electronic Service Associations, Wichita, Kansas, title: Transistor Fundamentals and Circuit Troubleshooting; **W. C. Caldwell,** supervisor, field service, before the Lafayette, Indiana, Rotary Club, title: Transistors and Electronics for the Future; **J. F. Martin,** field service engineer, before Kokomo, Indiana, High School electronics class, title: Transistor Fundamentals; **P. R. Powell,** field service engineer, before New Bedford, Massachusetts, School of Technology, title: Transistor Fundamentals; **J. R. Schaffner,** manager, semiconductor applications, before the National Cash Register Company engineer group, title: Design of Reliable Semiconductor Circuits; and **R. D. Stroup,** field service engineer, before the transportation branch, Tennessee Valley Association, Chattanooga, title: Transistor Fundamentals.

Guided Missiles and Space Technology

Bruce W. Davis, analytical staff, Defense Systems Division, before the American Rocket Society, Washington, D. C., title: Hard Versus Mobile Deployment of Intercontinental Ballistics Missiles.

From AC Spark Plug Division, Flint: **Leonard E. A. Batz,** design engineer, before Flint Optimist Club, title: How Rockets and Missiles Will Affect Our Daily Lives, and **G. B. Hardenbrook,** section manager—electronics, before the Frankenmuth, Michigan, Lion's Club, title: Missiles and Electronics.

From Allison Division: **J. L. Hartman,** assistant chief, Nuclear Engineering Department, before the central Indiana section, American Rocket Society, Indianapolis, title: Development of the Nuclear Rocket Engine, and **H. W. Welsh,** chief, advanced projects, before members of the U. S. Naval Reserve, Washington, D. C., title: Secondary Power Sources for Space Vehicles.

From AC Spark Plug Division's Milwaukee Operations: **James H. Bell,** director of guidance and navigation, before the American Society of Tool and Manufacturing Engineers, Milwaukee, title: Industry in the Missile Age; **Robert G. Brown,** director, ACRD—Milwaukee, before Milwaukee section, Institute of Radio Engineers, title: Feedback Controls for Inertially Guided Missiles; **L. A. Flaningam,** senior project engineer, before the Racine, Wisconsin, Optimist Club, title: Language of Missiles; **Anthony J. Italiano,** Mace program definition and control, before the American Society for Public Administration, Milwaukee, title: So Who Needs Pilots; **Victor O. Muth,** senior project engineer, before Oak Creek, Wisconsin, Parent Teachers Association, title: Missiles in Industry; and **Arnold B. Raninen,** director of Mace, before the American Society of Plant Engineers, Milwaukee, title: Inertial Guidance.

Highway and Traffic Engineering

From the GM Proving Grounds: **William G. Cichowski,** project engineer, before the 40th annual Highway Research Board meeting, Washington, D. C., title: Guardrail Installations—Appraisal by Proving Ground Car Impact and Laboratory Tests; **Louis C. Lundstrom,** director, before the American Bridge, Tunnel, and Turnpike Association, Albany, New York, title: Highway Safety; and **Kenneth A. Stonex,** assistant director, before the 9th annual construction conference, Cleveland Engineering Society, title: Roadside Design for Safety.

From the GM Research Laboratories: **D. C. Gazis,** senior research scientist, before the Northwestern University Traffic Engineering Seminar, title: Car Following Theories of Traffic Flow, and **Robert Herman,** head, Theoretical Physics, before The Ohio State University, graduate engineering lecture series, title: Theory of Traffic Flow.

Instrumentation

R. D. Tyler, superintendent, instrumentation and controls, Allison Division, before the Jet Testing Association, New York City, title: Instrumentation Techniques Used at Allison.

Paul C. Skeels, head, Experimental

Engineering Department, GM Proving Grounds, before the northeast Michigan section, Instrument Society of America, Midland, and before the Oak Ridge, Tennessee, section, I.S.A., title: Automotive Proving Ground Instrumentation.

Manufacturing

Olburtis Ball, tool engineer, Delco Radio Division, before the Metropolitan Kiwanis Club, Kokomo, Indiana, title: Epoxy Plastic Uses in Tooling.

Robert W. Chase, superintendent, product control testing, AC Spark Plug Division, before Flint, Michigan, Exchange Club, title: Reliability.

Claude V. Hawk, director of reliability, Harrison Radiator Division, before the annual management business meeting, American Society of Quality Control, Lockport, New York, title: Reliability Concept—Techniques of Application.

Edwin Pietrowicz, senior product engineer, Brown-Lipe-Chapin Division, before the Society of Die Cast Engineers, Syracuse, New York, title: Metal Flow Characteristics and Die Casting Dies.

From General Motors Institute: **Joseph O. McGinnis, Jr.**, faculty member, Production Engineering Department, before the central New York chapter, Society for the Advancement of Management, Syracuse, title: A New Concept of Methods Engineering; **Edward J. Polk**, faculty member, Production Engineering Department, before the western Michigan chapter, Industrial Management Society, Grand Rapids, title: Standardization and Analysis of Method and Time in Work Standards; and **Morris D. Thomas**, faculty member, Science Materials Department, and **Gordon L. Webster**, faculty member, Production Engineering Department, before the Grand Rapids, Michigan, section, American Society for Statistical Quality Control, title: Evolutionary Operation Applied to Resistance Welding of Automotive Sheet Metal.

From the GM Manufacturing Staff: **Robert F. Bentley**, department head—electronics, before a joint conference of the Michigan Industrial Arts Curriculum Committee and Teacher Educators, Pontiac, title: Progress in Metal Processing; **William A. Fletcher**, executive engineer, before the Saginaw Valley chapter, A.S.T.M.E., Flint, title: Cold Forming; and **David C. Salatin**, department head—metal casting, before the Columbus, Ohio, chapter, A.S.T.M.E., title: Evolution of Vacuum Die Casting.

Metallurgy

K. B. Valentine, metallurgical engineer, Pontiac Motor Division, before the Cleveland, Ohio, chapter, American Society for Metals, title: Influence of Heat Treatment on Engineering Properties of Steel, before the Canton-Massillon, Ohio, chapter, A.S.M., title: Some Materials Competitive With Steel, and before the York chapter, A.S.M. Harrisburg, Pennsylvania, title: Forgings, Castings, or Stampings.

A. L. Weiss, senior research engineer, Allison Division, before the Materials Advisory Board Symposium, National Academy of Sciences, Washington, D. C., title: Thermoelectric Generator Design With Materials That Exhibit Brittle Behavior.

From the GM Research Laboratories: **Manuel Ben**, supervisor, Electrochemistry Department, before the Newark, New Jersey, section, American Electroplaters Society, title: Decorative Plating on Aluminum; **W. L. Grube**, assistant head, Physics Department, and **S. R. Rouze**, senior research physicist, before the A.I.M.E. national metals congress, Philadelphia, title: Direct Observations on Twinning and Grain Growth in Austenite by Thermionic Emission Electron Microscopy; **T. J. Hugel**, supervisor, Metallurgical Engineering Department, before the Purdue University chapter, A.S.M., title: Thermoelectric Alloys — A New Area of Metallurgical Research; and before the Purdue University metallurgical engineering seminar, title: Precision Mechanical Properties of Beryllium for Gyro Applications; **H. A. Kahler**, liaison engineer, before the Detroit branch, A.E.S., title: Service Performance of Automotive Decorative Trim, and before the Grand Rapids, Michigan, branch, A.E.S., title: Decorative Plating for Automobiles; **J. G. Roberts**, senior research engineer, before the A.S.M.E., New York City, title: Residual Stresses and Metal Fatigue; and **R. F. Thomson**, head, Metallurgical Engineering Department, before the Hamilton, Ontario, branch, Canadian Institute of Mining and Metallurgy, title: Materials for the Automobile of the Future.

Research

Mieczykslaw G. Bekker, group head—land operations, Defense Systems Division, before the Scientific Advisory Committee, O.T.A.C., Detroit, title: Main Problems of Land Locomotion Research.

From the GM Research Laboratories: **H. M. Bendler**, research associate, before the Electron Diffraction Symposium, sponsored by the Franklin Institute of Philadelphia and the Philadelphia Electron Microscope Society, title: Electron Microscopy and Diffraction of Silicon Single Crystals; **Robert L. Dega**, supervisor, Mechanical Development Department, before the National Conference on Industrial Hydraulics, Chicago, title: Recent Advances in Lip Seal Technology; **R. C. Drutowski**, supervisor, Mechanical Development Department, before the joint A.S.M.E.-A.S.L.E. lubrication conference, Boston, title: The Volume of Stressed Material Involved in the Rolling of a Ball; **B. E. Nagel**, supervisor, Chemistry Department, before the American Association of Analytical Chemists, Detroit title: The Role of Analytical Chemistry in the Petroleum Laboratory; **H. A. Burley**, research engineer, and **Milton J. Diamond**, Central Foundry Division, before the Jackson, Michigan, chapter, A.S.M., title: A Fast Neutron System for Controlling the Moisture Content of Foundry Sand; **R. E. Marburger** and **D. P. Koistinen**, senior research physicists, before the American Society for Metals, Philadelphia, title: The Determination of Hardness in Steels from the Breadth of X-Ray Diffraction Lines; and **W. J. Mayer**, senior research chemist, **W. H. Lange**, research physicist, and **W. L. Shelly**, Allison Division, before the American Nuclear Society, San Francisco, title: The Measurement of AirFoil Wall Thicknesses of Hollow Turbine Blades and Vanes Using Sm-153.

GM Research Laboratories personnel who made presentations at the A.S.M.E. national meeting, New York City, included: **R. J. Moffat**, senior research engineer, title: Measurement of Air Velocity in a Smoke Tunnel by Means of Pulsed Smoke Streams; **Gino Sovran**, supervisor, Engineering Development Department, title: Rotating Stall in an Axial-Flow Fan and a Two Dimensional Cascade; **R. E. Black**, research physicist, title: Stimulating New Safe Industrial Uses of Radioisotopes; and **John L. Harned**,

senior research engineer, **P. Sudhindranath**, research engineer, and **K. M. Miller**, junior research engineer, title: Transfer Function Derivation for a New Departure Transitorq Variable Speed Drive.

GM Research Laboratories personnel who made presentations at the 8th American Association of Analytical Chemists conference, Wayne State University, Detroit, included: **Farno L. Green**, senior research physicist, title: Objectives of the Symposium on Applied Neutron Techniques of Analysis; **A. H. Jones**, senior research chemist, title: The Determination of Sodium, Lithium, and Calcium in Aluminum Alloys by Flame Photometry; **R. E. Kohn**, senior research chemist, and **Farno L. Green**, title: The Determination of Silicon in Cast Iron by Fast Neutron Activation Analysis; **R. B. Loranger**, research chemist, title: The Analysis of Cupola Slag by X-Ray Fluorescence; **B. E. Nagel**, supervisor, Chemistry Department, title: A Rapid Combustion Method for the Determination of Sulfur in the Presence of Chlorine; **A. C. Ottolini**, senior research chemist, title: The Application of X-Ray Fluorescence to the Analysis of Highly Alloyed Aluminum Samples; **T. P. Schreiber**, senior research physicist, title: Establishing and Controlling Analytical Curves; and **R. L. Gatrell**, research chemist, and **T. O. Morgan**, senior research chemist, title: Gas Chromatographic Determination of Carbon Dioxide and Water in Gaseous Samples.

Technical Careers and Vocational Guidance

R. W. Leland, staff engineer, Delco Products Division, before a high school career day program, Dayton, Ohio, title: Careers in Engineering.

Robert Budyak, engineer, **Merton Jacobs**, senior instructor—technical education, and **Richard Haertle**, supervisor—data processing, AC Spark Plug Division's Milwaukee Operations, before students of Dominican High School, Whitefish Bay, Wisconsin, titles: Career Opportunities in Electrical Engineering, Career Opportunities in Teaching, and Career Opportunities in Scientific Computation, respectively.

From AC Spark Plug Division, Flint: **Walter J. Sattler**, project engineer, before students of Flint Central High School, title: A Career in Electrical Engineering; **J. A. Stewart**, engineer, before students of Flint Southwestern High School, title: Electrical Engineering; and **D. E. Walz**, group leader, test engineering, before students of Flint Northern High School, title: Electrical Engineering.

Miscellaneous

Martin J. Caserio, general manager, Delco Radio Division, before the Evansville, Indiana, chapter, American Institute of Industrial Engineers, title: The Step Beyond the Next Step.

Richard E. Hallagan, supervisor, Engineering Illustration Department, Fisher Body Division, before the General Motors Institute Tech Club, Detroit, title: Engineering Illustration Today.

Richard O. Painter, head, Engineering Photography Department, GM Proving Grounds, before the southeastern Michigan section, Society for Experimental Stress Analysis, Dearborn, title: Equipment and Techniques for High Speed Photography.

Frank P. Rolf, supervisor, automotive testing, New Departure Division, before the City Educational Group, Terryville, Connecticut, title: Construction of the St. Lawrence Seaway.

From AC Spark Plug Division: **Robert W. Chase**, superintendent, product control testing, before students at Flint Northern High School, title: Memory Techniques in Studying and Learning, and **Harold S. Sharp**, technical librarian, Milwaukee Operations, before the Instrument Society of America's winter instrument-automation conference, St. Louis, title: The Technical Library as a Tool of Engineering.

From Allison Division: **C. B. Butts**, test engineer, before the Air Transport Association, Washington, D. C., title: Non-Destructive Testing of Aircraft Engine Components at Allison; **G. J. Clingman**, senior service engineer, before the Indiana Home Office Underwriters, Indianapolis, title: The Electra Story; and **M. E. Fisher**, senior project engineer, before the National Conference on Industrial Hydraulics, Chicago, title: Hydraulic Control System for a Track-Type Tractor Power Shift Transmission.

From General Motors Institute: **Galen H. Frantz**, faculty member, Management Training Department, before students of Manchester College, Manchester, Indiana, title: Our Approach on Group Dynamics; **S. A. Smith**, faculty member, Management Training Department, before the Detroit chapter, Personnel Management Association, title: Principles of Management Development; and **W. R. Struwin**, faculty member, Science-Energy Department, before Michigan section, American Association of Physics Teachers, Detroit, title: Laboratory Determination of Gamma for Gases.

From Guide Lamp Division: **H. E. Welker**, senior process engineer, before the Indianapolis branch, American Electroplaters Society, title: How to Train Foremen and Operators in the Most Efficient Application of Buff and Compound. Before a group of engineering educators from Purdue University: **E. W. Brock**, **J. H. Diedring**, and **L. N. Williams**, staff engineers, and **J. W. Murphy**, engineer, general title: Why Do We Need Your Product, with specific discussion on electronics, design, materials, and optics, respectively.

From GM Engineering Staff: **William H. Kolbe**, section engineer, before the American Society of Heating, Refrigerating, and Air Conditioning Engineers, Tulsa, Oklahoma, title: Design of an Axial Refrigeration Compressor; **Roy P. Trowbridge**, director, engineering standards, before the American Society for Testing Materials, Detroit, title: What Makes a Good Specification; and **Stephen Kalmar**, executive assistant engineer in charge, Power Development Group, and **William K. Steinhagen**, assistant engineer in charge, Power Development Group, before students at Michigan State University—Oakland, titles: Engineering Administration Procedure and Technical Supervision—Project Coordination, respectively.

From GM Research Laboratories: **John M. Campbell**, scientific director, before the Conference on Management of Research, University of Wisconsin, title: Communications with Management, and **W. D. Cheek**, senior research physicist, before the Jackson, Michigan, chapter, American Society for Metals, title: Fission, Fusion, and Confusion.

A Typical Problem in Process Engineering:

Develop a Process Plan to Control Tolerance Stack Conditions for a Machined Part

By ROBERT DINDA
and L. C. LANDER, JR.
General Motors Institute

Various forms of process planning are applied in the fabrication of automotive, appliance, and aircraft components. A very important phase of process planning is to make a detailed dimensional analysis of the part to be manufactured to assure that there will not be excessive tolerance stack conditions. When making such a dimensional analysis, it also is important to understand the capabilities of various processing operations so that optimum manufacturing costs will be realized. The problem presented here is to develop a process plan which will facilitate limit stack control and optimum selection of tools and equipment needed for the manufacture of a machined part.

A part print

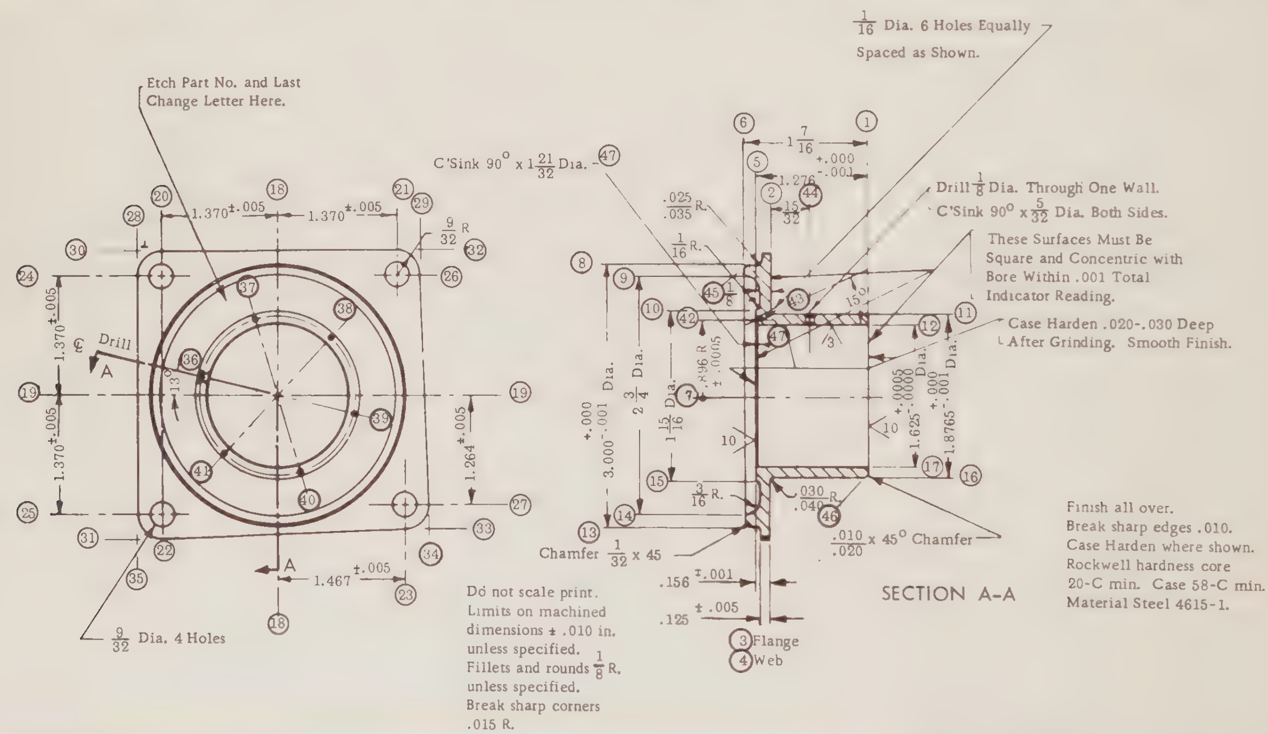
is the only

given data

THE PROCESS planning of machined parts invariably results in the issuance of the *process routing*. The process routing summarizes the manufacturing operations to be performed, the sequence in which they are to be performed, and the tools required.

The procedure normally followed in planning a process is to determine first what manufacturing processes are required. The requirements are determined from an analysis of engineering specifications, usually in the form of a part print (Fig. 1). Alternative manufacturing processes which might be applicable are then considered and the optimum process is selected from the standpoint of equipment availability and economics. Finally, the operational requirements are determined for the selected process.

The next step is to determine what surfaces on the workpiece are to be used for locating and measuring during the various processing operations. To some



The problem presented here is typical of problems assigned to students enrolled in a manufacturing processing course at General Motors Institute. Data for the problem is derived from an inspection and gaging problem included in the course text. The problem has been expanded, however, to provide the student the opportunity to make a detailed analysis of the factors involved in process and machine selection.

Fig. 1—This is a part print of a bearing cage. A process plan is to be developed for its manufacture. The encircled numbers, shown in color, are code numbers for each surface of the bearing cage. Each dimension can be easily identified by noting the code number of the surfaces between which the dimension is located. The use of code numbers provides a handy means of reference when making the required detailed dimensional analysis to control tolerance stack conditions.

extent, these surfaces establish the initial processing operations in the total operational sequence. Secondary and allied operations are established and the total operational requirements and sequence are completed.

The selection of tools and equipment to perform each operation ends the primary activity of the process engineer. Effective tool and equipment selection requires the compilation of sufficient quantitative information to facilitate the matching of process requirements with tool and equipment capabilities.

Problem

The problem is to develop a process plan for the manufacture of a machined part called a bearing cage (Fig. 1). A detailed dimensional analysis should be conducted and arranged in a suitable form which will facilitate the control of tolerance stack conditions and the selection of tools and equipment. The actual selection of the *specific* tools and equipment to use is not to be a part of this problem. The processing operations to be performed, however, such as machining, grinding, and drilling, should be specified as part of the overall process plan. The intent of the problem is to develop that information which will be of value in selecting the required tools and equipment.

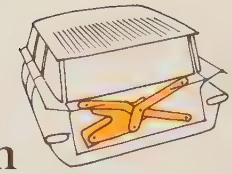
The solution to the problem will be presented in the July-August-September 1961 issue of the General Motors Engineering Journal.

New Booklet on Automotive Engineering Available to Educators

Some of the past and present efforts of General Motors people—the inventors, scientists, and engineers—and some of their accomplishments in the field of automotive transportation is described in a booklet entitled "A Better Way—A Story of General Motors Automotive Engineering." Copies may be obtained by writing to Public Relations Staff, General Motors Corporation, General Motors Building, Detroit 2, Michigan.

Solution to the Previous Problem:

Analyze the Mechanical Components of an Automotive Window Regulator Mechanism



The mechanical components of an automotive window regulator mechanism consist of stampings joined together by pins and rivets. These components must have sufficient strength to meet specific functional requirements—namely, the ability to withstand bending moments and shear loads resulting from the application of a test load torque of 250 in-lb applied at a spindle-mounted pinion. This is the solution to the problem presented in the January-February-March 1961 issue of the General Motors Engineering Journal. The problem required that a stress analysis be made on various components of the regulator mechanism to see if they were structurally adequate to withstand the applied loads. The problem also required that a torsion-type counterbalance spring be designed which would be capable of balancing the window glass assembly when it is in the fully lowered position.

By JOHN P. BURLEY
Fisher Body Division

Assisted by E. H. Eddy
General Motors Institute

Stress analysis shows some design changes are necessary

The functional test of the automotive window regulator mechanism (Fig. 1) consists of applying a torque of 250 in-lb to the spindle-mounted pinion while the lift pins at each end of the balance arm and lift arm are restrained against movement. Under these load conditions the components of the mechanism must be checked for structural adequacy. Before this can be done, however, the influence that the counterbalance spring (Fig. 2) has on the functional test loads must first be determined.

The counterbalance spring must be capable of completely balancing the weight of the glass assembly when it is in the fully lowered position. This means that the torque output of the spring must be sufficient to counterbalance the external moment of the window glass assembly when it is in the fully lowered position.

The external moment M_E of the window glass assembly is equal to the product of the glass assembly weight and the lever arm from the center of gravity of the glass assembly to the main pivot point (point A, Fig. 1) of the regulator mechanism, plus the product of the reaction at the fixed channel end of the balance arm (point F) and its lever arm to the main pivot point. The external moment M_E, therefore, is equal to

$$M_E = (23.3)\,(9.18) + \left[11.65\left(\frac{13.19}{9.89}\right)\right]19.78$$

$$M_E = 520 \text{ in-lb.}$$

The counterbalance spring can be investigated as a flat wire, torsion-type spring design. The fundamental formulas for a spring of this type are:

$$S = \frac{6M}{bh^2} \qquad (1)^*$$

$$N = \frac{SL}{\pi Eh} \qquad (2)$$

where

S = stress (psi)
M = torque (in-lb)
b = material width—axial (in.)
h = material thickness—radial (in.)
N = number of turns spring will give
L = length of active material in spring (in.)
E = modulus of elasticity of spring material (psi).

*The straight beam formula (equation 1) is applied as standard practice. After the coiling process for a torsion-type spring is completed, the outer surface of each coil is in compression. The apparent stress to which torsion springs can be subjected without set may be greater than the proportional limit in bending of a straight piece of the same stock from which they were made.

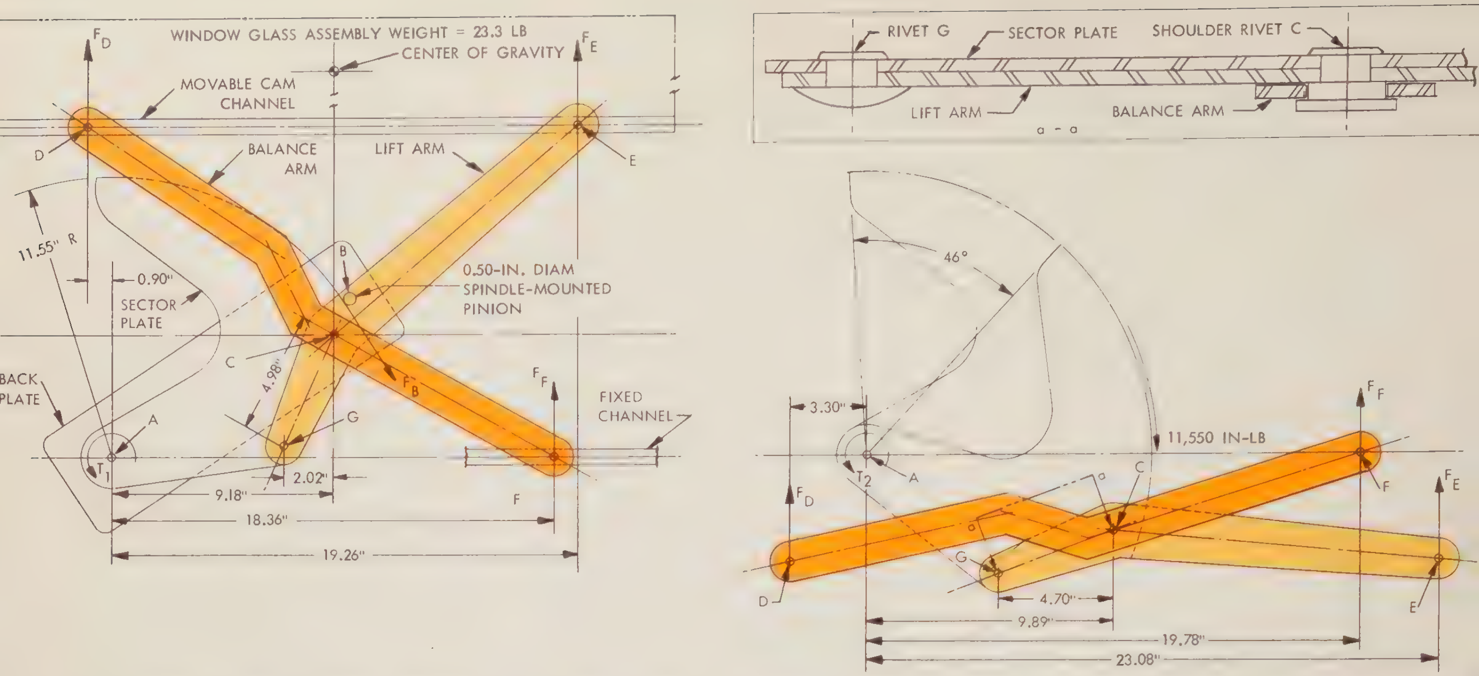


Fig. 1—An automotive window regulator mechanism, such as might be used in the tailgate window of a station wagon, is shown here in the fully raised position (left) and the fully lowered position (right). The main components of the window regulator mechanism are a sector plate, a back plate, a spindle-mounted pinion, a lift arm, and a balance arm. Rivets and pins are used to connect the main components.

The sector plate pivot pin is located at point *A*. The full angular travel of the sector plate between the raised and lowered positions is 46°. Also located at point *A* is a torsion-type counterbalance spring (Fig. 2). The curved vector T_1 represents the output torque of the spring when the window is fully raised. When the window is fully lowered the output torque of the spring is represented by the curved vector T_2. A test load torque of 250 in-lb applied to the spindle-mounted pinion, point *B*, produces a moment of 11,550 in-lb around point *A* and a tangential force F_B tangent to the point of engagement between the pinion *B* and the sector plate.

The balance arm has a center pivot pin at point *C*. This pin also serves to rivet the lift arm to the sector plate. The lift arm also is riveted to the sector plate at point *G*. The balance arm has a lift pin roller attached to a movable cam channel at point *D*. The lift arm has a similar pin roller attached to a movable cam channel at point *E*. The lower end of the balance arm is attached to a fixed channel by a pin roller at point *F*.

The 250 in-lb test load produces vertical reactions F_D, F_E, and F_F at points *D*, *E*, and *F*. These points are loaded in the vertical direction only because the pin rollers are free to move horizontally in the channels. The relationship between lift pins *D* and *E*, with respect to their distances from pivot pin *C*, is held symmetrical by the equal distance of points *A* and *F* to point *C* and by the common horizontal centerline of points *A* and *F*.

The desired coil width (axial) of the counterbalance spring was given as 0.50 in. Substituting this value and the calculated value for the external moment into equation (1) gives

$$S = \frac{6(520)}{(0.50)h^2} = \frac{6{,}240}{h^2}$$

or,

$$h^2 = \frac{6{,}240}{S}.$$

For a carbon steel spring material of S.A.E. 1050-1085, the elastic limit for tensile strength is from 130,000 psi to 240,000 psi[1]. Substituting a mean value of 185,000 psi as the elastic limit into the preceding equation for h^2 gives

$$h^2 = \frac{6{,}240}{185{,}000} = 0.0337$$

$$h = 0.184 \text{ in.}$$

The standard stock thickness which is nearest to the calculated thickness is 0.179 in. The maximum stress, therefore, to which the spring will be subjected is

$$S_{max} = \frac{6{,}240}{0.032} = 195{,}000 \text{ psi.}$$

This stress is within the elastic limit range for the S.A.E. carbon steel.

To determine if the required actual length for a flat wire, torsion-type spring having the width and thickness just determined is within the desired maximum coil length of 65 in., it is necessary to find the number of turns that this spring will have to travel. The desired spring rate was given as 2 in-lb per degree. The total number of turns *N*, therefore, can be calculated as follows:

$$N = (520 \text{ in-lb}) \left(\frac{1°}{2 \text{ in-lb}}\right) \left(\frac{1 \text{ turn}}{360°}\right)$$

$$= 0.72 \text{ turns.}$$

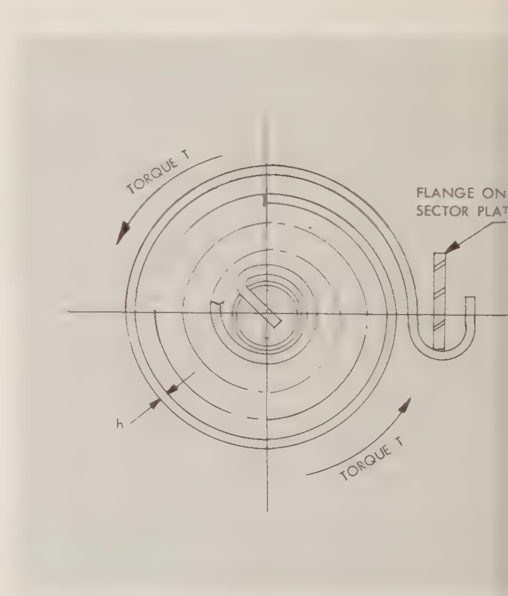


Fig. 2—A flat wire, torsion-type counterbalance spring of thickness *h* is used in the window regulator mechanism. The spring, which is keyed to the back plate, is located at the main pivot pin on the sector plate (point *A*, Fig. 1). The moving end of the spring is fastened to a flange on the sector plate.

Substituting this value for N into equation (2) gives

$$0.72 = \frac{(195{,}000)\ (L)}{(\pi)\ (30)\ (10^6)\ (0.179)}$$

$$L = 62.3 \text{ in.}$$

This length is below the desired maximum spring length of 65 in. The spring design, therefore, is acceptable.

The angular travel of the sector plate between the fully raised and fully lowered positions of the regulator mechanism is 46° (Fig. 1-right). The output torque T_1 of the counterbalance spring when the window is in the fully raised position, therefore, is equal to

$$520 - 46° \left(\frac{2 \text{ in-lb}}{1°}\right) = 428 \text{ in-lb.}$$

The output torque T_2 of the counterbalance spring when the window is in the fully lowered position is 520 in-lb, which is the same as the external moment.

Forces on Regulator Mechanism Lift Pins Analyzed

The first step in analyzing the various components of the regulator mechanism is to determine the forces at the lift pins D and E (Fig. 1) when the regulator is in the fully raised position.

When the test load torque of 250 in-lb is applied to the spindle in a counterclockwise direction, the tangential force F_B is equal to

$$\frac{250}{0.25} = 1{,}000 \text{ lb.}$$

Taking moments about point A (Fig. 1-left):

$$M_A = (1{,}000)\ (11.55) - T_1 + (F_D)\ (0.90) - (F_E)\ (19.26) - (F_F)\ (18.36) = 0.$$

Since the balance arm and the lift arm are both commonly restrained in a vertical direction only, it can be assumed that the restraining forces F_D and F_E are equal.

Taking moments about point C on the balance arm:

$$(F_D)\ (10.08) - (F_F)\ (9.18) = 0$$

$$F_F = 1.10F_D.$$

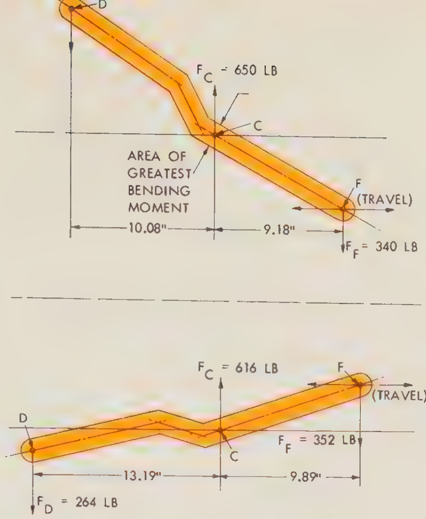


Fig. 3—Shown here are free body diagrams of the balance arm when it is in the fully raised position (top) and the fully lowered position (bottom).

Substituting this value for F_F into the preceding equation for moments taken about point A gives

$$M_A = 11{,}550 - 428 + 0.90F_D - 19.26F_D - 18.36(1.10F_D) = 0$$

$$F_D = 290 \text{ lb.}$$

The vertical reactions, therefore, are:

$$F_D = 290 \text{ lb}$$
$$F_E = 290 \text{ lb}$$
$$F_F = 319 \text{ lb.}$$

Applying the test load torque of 250 in-lb in a clockwise direction and taking moments about point A will give the following values for forces F_D, F_E, and F_F:

$$F_D = F_E = 310 \text{ lb}$$
$$F_F = 340 \text{ lb.}$$

These forces, which are in a direction opposite to the vectors shown in Fig. 1-left, produce the greater bending moments when the regulator mechanism is in the fully raised position.

The next step in the analysis is to determine the lift pin forces F_D and F_E when the regulator mechanism is in the fully lowered position (Fig. 1-right). Applying the test load torque in a counterclockwise direction and taking moments about point A gives

$$M_A = 11{,}550 - T_2 + (F_D)\ (3.30) - (F_E)\ (23.08) - (F_F)\ (19.78) = 0.$$

Taking moments about point C on the balance arm:

$$F_F = \frac{13.19}{9.89} = 1.33F_D.$$

Using the assumption that F_D equals F_E and substituting into the equation for moments about point A gives

$$M_A = 11{,}550 - 520 + 3.30(F_D) - 23.08(F_D) - 19.78(1.33F_D) = 0$$

$$46.15F_D = 11{,}030$$
$$F_D = 240 \text{ lb}$$
$$F_E = 240 \text{ lb}$$
$$F_F = 318 \text{ lb}$$

These forces are in the direction shown in Fig. 1-right.

Applying the test load torque of 250 in-lb in a clockwise direction and resolving moments gives

$$F_D = F_E = 264 \text{ lb}$$
$$F_F = 352 \text{ lb.}$$

These forces, which are in a direction opposite to the vectors shown in Fig. 1-right, produce the greater bending moments when the regulator mechanism is in the fully lowered position.

Free Body Diagrams of Link Members Analyzed

The next step in the solution is to analyze the balance arm and link arm members of the regulator mechanism by means of free body diagrams.

Balance Arm

The free body diagram for the balance arm (Fig. 3) treats the arm as a simple beam having its fulcrum at point C and loaded at points D and F.

A summation of forces in the vertical direction for the balance arm when it is in the fully raised position (Fig. 3-top) gives:

$$F_D + F_C + F_F = 0$$
$$F_D = 310 \text{ lb at } 270°$$
$$F_F = 340 \text{ lb at } 270°$$

Therefore,

$$F_C = 650 \text{ lb at } 90°.$$

The maximum moment, which is at point C, is equal to

$$M_C = (F_D)\ (10.08)$$
$$M_C = (310)\ (10.08) = 3{,}130 \text{ in-lb.}$$

The same procedure can be followed to determine F_C and the maximum moment at point C when the balance arm is

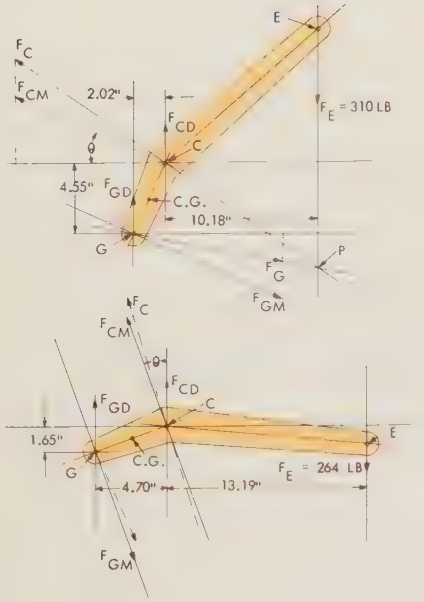


Fig. 4—These are free body diagrams of the lift arm when in the fully raised position (top) and fully lowered position (bottom). The arm is treated as a cantilever beam loaded at point *E*. Reactions are at points *C* and *G*. The centroid of the lift arm is shown as point *C.G.*

in the fully lowered position (Fig. 3-bottom). These values are:

$$F_C = 616 \text{ lb}$$
$$M_C = 3{,}480 \text{ in-lb.}$$

This analysis shows that the balance arm is subjected to the greatest bending moment when it is in the fully lowered position. Also, the lift pin *F* is subjected to the greatest shear load when the arm is in the fully lowered position.

The next step is to check the bending stress at point *C* by applying the fundamental relationship

$$S = \frac{Mc}{I} \qquad (3)$$

where

S = bending stress (psi)

M = maximum bending moment (in-lb)

c = distance from neutral axis of balance arm to remotest fiber from neutral axis (in.)

I = moment of inertia about neutral axis (in.)

The thickness *b* of the balance arm was given as 0.120 in. and the width *h* (height) as 1.65 in. The diameter of the hole at point *C* is 0.80 in. Therefore,

$$c = \frac{1.65}{2} = 0.825 \text{ in.}$$

$$I = \frac{bh^3}{12} = \frac{(0.120)\ (1.65^3)}{12} - \frac{(0.120)\ (0.80^3)}{12}$$

$$I = 0.0398 \text{ in.}^4$$

Substituting known values for *M*, *c*, and *I* into equation (3) gives the following value for bending stress:

$$S = \frac{(3{,}480)\ (0.825)}{0.0398} = 72{,}200 \text{ psi.}$$

This stress exceeds the elastic limit for the S.A.E. 1010 steel selected as the balance arm material.

If the thickness of the balance arm is increased to 0.135 in., the bending stress is reduced to

$$S = \frac{0.120}{0.135}\,(72{,}200) = 64{,}200 \text{ psi.}$$

This stress is still greater than the elastic limit. Any further increase in thickness would penalize the overall balance arm for strength at point *C*.

Letting the bending stress equal 32,500 psi and the thickness equal 0.135 in., the required section modulus *Z* at point *C* of the balance arm would be

$$Z = \frac{M}{S} = \frac{3{,}480}{32{,}500} = 0.107 \text{ in.}^3$$

If the height *h* is increased to 2.25 in. at point *C*, the section modulus *Z* therefore is equal to

$$Z = \frac{0.135}{6}\left(\frac{2.25^3 - 0.51}{2.25}\right)$$

$$Z = 0.109 \text{ in.}^3$$

This section modulus is acceptable for the conditions at point *C*. Since the bending moment is zero at the ends, the height of the balance arm can be tapered from 2.25 in. at the center around point *C* to 1.65 in. at each end.

The pin diameters at points *C*, *D*, and *F* must be checked to see if they are adequate.

Stress at point *C*:

Minimum pin diameter = 0.50 in.

Maximum load = 650 lb

$$S = \frac{4(650)}{\pi(0.50^2)} = 3{,}300 \text{ psi.}$$

Stress at point *D*:

Pin diameter = 0.25 in.

Maximum load = 310 lb

$$S = \frac{4(310)}{\pi(0.25^2)} = 6{,}320 \text{ psi.}$$

Stress at point *F*:

Pin diameter = 0.38 in.

Maximum load = 352 lb

$$S = \frac{4(352)}{\pi(0.38^2)} = 3{,}100 \text{ psi.}$$

The three pins at points *C*, *D*, and *F* have adequate diameters, since the calculated stresses are all within the elastic shear limit of 36,000 psi for S.A.E. 1030 cold rolled steel.

Lift Arm

The free body diagram for the lift arm (Fig. 4) treats the arm as a cantilever beam loaded at point *E* and having reactions at points *C* and *G*.

Points *C* and *G* can be considered as an eccentrically loaded rivet pattern. The center of resistance of the pattern is located at its centroid *C.G.* Using this approach, the direct loads F_{CD} and F_{GD} and the moment loads F_{CM} and F_{GM} can be easily determined.

The moment loads can be determined by applying the following formula:

$$F_{N1} = \frac{MR_{N1}}{(R_{N1})^2 + (R_{N2})^2 + \ldots . (R_{Nx})^2}$$

where

F_{N1} = force on rivet N_1

M = total moment on rivet pattern

R_{N1} = distance from rivet N_1 to centroid

R_{N2} = distance from rivet N_2 to centroid

R_{Nx} = distance from rivet N_x to centroid.

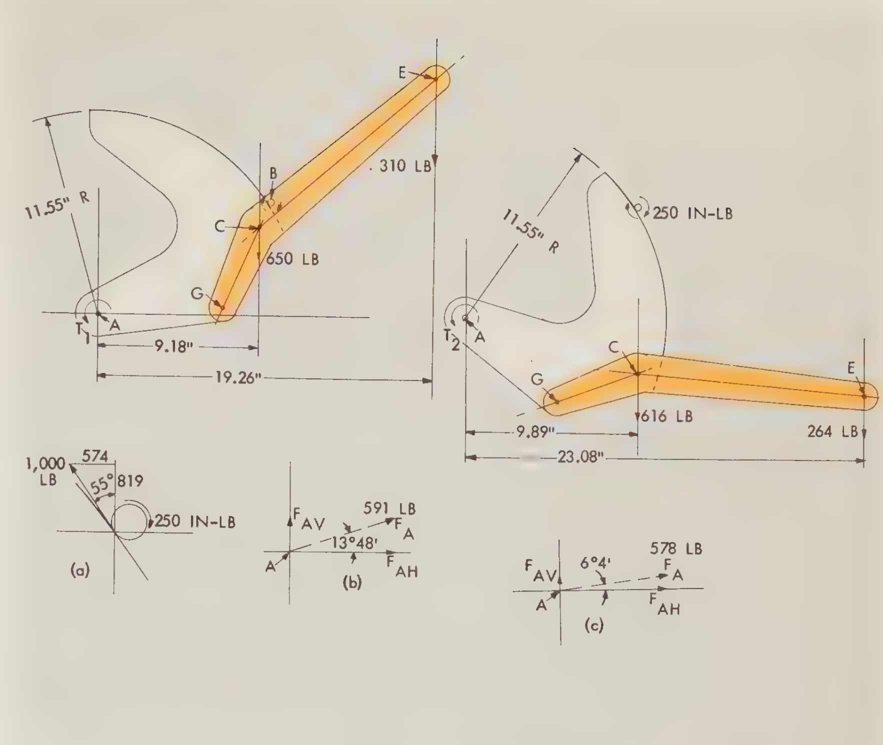
When the lift arm is in the fully raised position (Fig. 4-top) the total moment on the rivet pattern is equal to

$$M = (310)(11.09) = 3{,}438 \text{ in-lb.}$$

Since there are only two rivets,

$$R_{NC} = R_{NG} = 2.49 \text{ in.}$$

Fig. 5—The lift arm and sector plate assembly is subjected to the same two external loads when it is in either the fully raised position (left) or the fully lowered position (right). The first load is a 1,000-lb load acting at the tangential point of engagement between the pinion B and the sector plate. This load results from the test load torque of 250 in-lb applied at the pinion (a). The second external load is the counterbalance spring output torque of 428 in-lb around point A. The assembly also is subjected to internal loads from the balance arm and lift arm at points C and E. To determine the reactions at point A, the horizontal and vertical components of the 1,000 lb tangential force must first be found. The resultant force at point A is then calculated. When the assembly is in the fully raised position, the resultant force F_A is equal to 591 lb at 13° 48′ (b). When the assembly is in the fully lowered position, the resultant force F_A is equal to 578 lb at 6° 4′ (c).



Therefore,

$$F_{CM} = \frac{(3{,}438)(2.49)}{(2.49)^2 + (2.49)^2}$$

$$F_{CM} = 690 \text{ lb at } (180 - \Theta)°$$

and

$$F_{GM} = 690 \text{ lb at } (360 - \Theta)°$$

where

$$\tan \Theta = \frac{2.02}{4.55} = 0.44396$$

$$\Theta = 23° \; 56' \; 20''.$$

The direct loads F_{CD} and F_{GD} are equal to one-half the total applied load of 310 lb at point E. Therefore,

$$F_{CD} = 155 \text{ lb at } 90°$$

$$F_{GD} = 155 \text{ lb at } 90°.$$

The force F_C can be found by first determining the horizontal and vertical vectors of F_{CM} and then adding F_{CD} vectorially as follows:

$$F_{CM} = 690 \text{ lb at } 156° \; 4'$$

or

$$F_{CM} = 280 \text{ lb at } 90° \nrightarrow 630.6 \text{ lb at } 180°$$

$$+F_{CD} = 155 \text{ lb at } 90° \nrightarrow 0.$$

Therefore,

$$F_C = 435 \text{ lb at } 90° \nrightarrow 630.6 \text{ lb at } 180°.$$

Resolving:

$$F_C = 766 \text{ lb at } 145° \; 24'.$$

The same procedure is followed to find the force F_G.

$$F_{GM} = 690 \text{ lb at } 336° \; 4'$$

or

$$F_{GM} = 280 \text{ lb at } 270° \nrightarrow 630.6 \text{ lb at } 0°$$

$$+F_{GD} = 155 \text{ lb at } 90° \nrightarrow 0.$$

Therefore,

$$F_G = 125 \text{ lb at } 270° \nrightarrow 630.6 \text{ lb at } 0°.$$

Resolving:

$$F_G = 642 \text{ lb at } 348° \; 74'.$$

Extending forces F_C, F_E, and F_G in their respective vector directions shows that they are concurrent forces intersecting at point P (Fig. 4-top). This fulfills the theorem of three, non-parallel forces in equilibrium which states that when three, non-parallel forces are in equilibrium, they are concurrent and coplanar[2].

The same method of analysis can be applied to determine forces F_C, F_E, and F_G when the lift arm is in the fully lowered position (Fig. 4-bottom). These forces are as follows:

$$F_C = 950 \text{ lb at } 106° \; 58'$$

$$F_E = 264 \text{ lb at } 270°$$

$$F_G = 701 \text{ lb at } 293° \; 17'.$$

These three forces also meet the requirements of the theorem of three non-parallel forces in equilibrium.

The lift arm has a maximum bending moment, when in the fully raised position, of

$$M = (310)(10.08) = 3{,}125 \text{ in-lb}.$$

When the lift arm is in the fully lowered position, the maximum bending moment is

$$M = (264)(13.19) = 3{,}482 \text{ in-lb}.$$

The maximum bending stress in the lift arm, therefore, is equal to

$$S = \frac{Mc}{I}$$

where

$$M = 3{,}482 \text{ in-lb}$$

$$c = 1.10 \text{ in.}$$

$$I = I_{arm} - I_{hole}$$

$$I = \frac{(0.120)(2.20^3)}{12} - \frac{0.120(0.80^3)}{12}$$

$$I = 0.1014 \text{ in.}^4$$

$$S = \frac{(3{,}482)(1.10)}{0.1014} = 37{,}800 \text{ psi}.$$

This bending stress exceeds the elastic limit of 32,500 psi for the S.A.E. 1010 steel selected as the lift arm material. Increasing the thickness of the lift arm to 0.140 in. gives the following value for bending stress:

$$S = \frac{0.120}{0.140}(37{,}800) = 32{,}400 \text{ psi}.$$

This stress is within the elastic limit of S.A.E. 1010 steel.

The stress in the pins and rivets at points C, E, and G must now be checked.

Stress at Point C:

Rivet diameter = 0.50 in.

Maximum load = 950 lb at 106° 58′ ⧺ 616 lb at 90°

Maximum load = 1,550 lb at 100° 19′

$$S = \frac{4(1{,}550)}{(\pi)(0.50^2)} = 7{,}890 \text{ psi.}$$

Stress at point E:

Pin diameter = 0.25 in.

Maximum load = 310 lb

$$S = \frac{4(310)}{\pi(0.25)^2} = 6{,}320 \text{ psi.}$$

Stress at point G:

Rivet diameter = 0.50 in.

Maximum load = 701 lb

$$S = \frac{4(701)}{\pi(0.50)^2} = 3{,}560 \text{ psi.}$$

These calculations show that the stresses at points C, E, and G are acceptable.

Lift Arm—Sector Plate Analyzed

When the lift arm—sector plate assembly is in the fully raised position (Fig. 5-left) the sector plate receives two external loads: (a) the 250 in-lb test load, in the form of a 1,000 lb load at the tangential point of engagement between the pinion B and the sector plate, and (b) the counterbalance spring output torque T_1 of 428 in-lb around point A.

A summation of the external moment acting on the lift arm—sector plate assembly gives:

(1,000)(11.55) + 428 = 11,978 in-lb in a counterclockwise direction.

The assembly also is subjected to internal loads from the balance arm and lift arm at points C and E. These loads are as follows:

At point C:
650 lb at 270°

At point E:
310 lb at 270°.

The summation of the internal moments is:

(650) (9.18) + (310) (19.26) = 11,938 in-lb in a clockwise direction.

To determine the reactions at point A, the horizontal and vertical components of the 1,000 lb tangential force must be found. The angle between the horizontal and vertical components is 55° (Fig. 5a). The component forces, therefore, are equal to:

Horizontal Component:

(1,000) (cos 55°) = 574 lb at 180°

Vertical Component:

(1,000) (sin 55°) = 819 lb at 90°.

A summation of all vertical components gives:

819 lb at 90° ⧺ 650 lb at 270°

⧺ 310 lb at 270° ⧺ F_{AV} = 0

F_{AV} = 141 lb at 90°.

A summation of all horizontal components gives:

574 lb at 180° ⧺ F_{AH} = 0

F_{AH} = 574 lb at 0°.

The resultant force at point A (Fig. 5b) when the lift arm—sector plate assembly is in the fully raised position, therefore, is equal to:

F_A = F_{AV} ⧺ F_{AH}

F_A = 141 lb at 90° ⧺ 574 lb at 0°

F_A = 591 lb at 13° 48′.

The stress on the pin at point A is equal to:

$$S = \frac{P}{A} = \frac{4(591)}{\pi(0.50)^2}$$

S = 3,000 psi.

This stress is within the elastic limit for S.A.E. 1030 cold rolled steel.

The external loads acting on the lift arm—sector plate assembly when in the fully lowered position (Fig. 5-right) are the same as those acting on the assembly when it is in the fully raised position. The internal loads from the balance arm and the lift arm are:

At point C:

616 lb at 270°

At point E:
264 lb at 270°.

A summation of the internal moments gives:

(616) (9.89) + (264) (20.38) = 12,185 in-lb.

To determine the force at point A, the vertical components are first summarized as follows:

819 lb at 90° ⧺ 616 lb at 270°

⧺ 264 lb at 270° ⧺ F_{AV} = 0.

F_{AV} = 61 lb at 90°.

A summary of all horizontal components gives:

574 lb at 180° ⧺ F_{AH} = 0

F_{AH} = 574 lb at 0°.

The resultant force at point A, therefore, when the lift arm—sector plate assembly is in the fully lowered position (Fig. 5c) is:

F_A = 61 lb at 90° ⧺ 574 lb at 0°

F_A = 578 lb at 6° 4′.

The stress on the pin at point A is equal to:

$$S = \frac{4(578)}{\pi(0.50)^2} = 3{,}000 \text{ psi.}$$

This stress is acceptable.

Summary

A summary of the analysis made on the window regulator mechanism in regard to its ability to meet the required functional specifications is as follows.

Counterbalance Spring

The counterbalance spring should be a flat wire, torsion-type spring having the following specifications:

Material—S.A.E. 1050-1085

Minimum yield strength—200,000 psi

Minimum ultimate strength—250,000 psi

Coil Width (Axial)—0.50 in.

Coil Thickness (Radial)—0.179 in.

Spring Length (Active)—62.3 in.

Balance Arm

The thickness of the balance arm should be increased from 0.120 in. to 0.135 in. The width of the arm should be

increased to 2.25 in. at the center and should taper to a width of 1.65 in. at both ends.

Lift Arm

The thickness of the lift arm should be increased from 0.120 in. to 0.140 in.

Sector Plate

No changes in the original design of the sector plate are recommended.

Pin and Rivet Connections

The pivot pins and rivets are structurally adequate. Although the stresses are low, it is not recommended that the diameters be reduced because of the planar rigidity that the larger diameters provide between the inter-connected components.

Bibliography

1. "Manual of Spring Engineering," published by the American Steel and Wire Division, United States Steel Corporation, August 1958.
2. HIGDON, A., and STILES, W. B., *Engineering Mechanics* (New York: Prentice-Hall, Inc., 1948), p. 88.

Reprints of Patent Articles Available to Educators

Since the first issue of the GENERAL MOTORS ENGINEERING JOURNAL, members of the GM Patent Section have contributed articles dealing with the more generally encountered aspects of patents, patent law, and related matters. These articles have been reprinted in a separate booklet entitled "Inventions, Patents, and Related Matters." The booklet is intended to serve as a convenient source of supplementary information for educators who may be concerned with patent law matters in connection with classroom work. These educators may request a copy from the Educational Relations Section, Public Relations Staff, General Motors Corporation, GM Technical Center, Warren, Michigan.

Contributors to April-May-June 1961 Issue of GENERAL MOTORS ENGINEERING JOURNAL

JOHN P. BURLEY, contributor of the problem, "Analyze the Mechanical Components of an Automotive Window Regulator Mechanism," and the solution appearing in this issue, is a stress analyst with the Body Engineering Activities of Fisher Body Division. His present responsibilities include the structural analyses of tailgate and liftgate mechanical components and convertible top mechanisms. He also calculates the section properties of front end sheet metal structural members.

Mr. Burley joined Fisher Body in 1952 as a General Motors Institute student in the cooperative engineering program. He received a B.M.E. degree from G.M.I. in 1957. Prior to assuming his present position, he was a layout draftsman in the Design and Drafting Department at Fisher Body.

Mr. Burley is a member of the Society of Automotive Engineers and the Engineering Society of Detroit. He is presently doing graduate study at the University of Detroit.

WARD D. HALVERSON, co-contributor of "Analyzing the Exhaust Velocity Requirements for Electrically Powered Rockets," is a research engineer in the Advanced Propulsion Devices Section of Allison Division's Engineering Research Department. He is currently concerned with theoretical studies on the problem of charge neutralization in the exhaust beam of electrostatic space rockets.

Mr. Halverson was granted a B.S. (Geophysics) degree from the Massachusetts Institute of Technology in 1956. He then was awarded a one year fellowship to study geophysics at the Institut Francais du Petrole. Upon return to this country in 1957, he joined Allison Division as an engineer-in-training.

Mr. Halverson's previous major projects have included space trajectory analysis, space mission requirement studies, and magnetofluiddynamics research. He is a member of the American Rocket Society and has been the co-author of two published papers dealing with the electrogasdynamic analysis of ion jet neutralization.

EARL E, LINDBERG contributor of "A Discussion on the Out-of-Roundness of Machined Parts and its Measurement," is a senior research engineer in the Mechanical Development Department of the General Motors Research Laboratories.

The Mechanical Development Department is concerned with a variety of projects such as unconventional engines, Diesel engines, fatigue testing, friction, lubricants, and bearings. A considerable portion of Mr. Lindberg's work has been related to the development of electronic instrumentation for this Department.

Mr. Lindberg received the Bachelor of Science degree in electrical engineering from the University of Nebraska in 1949. He joined General Motors in 1951.

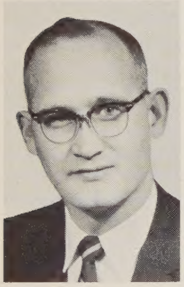
He is a member of the Institute of Radio Engineers and the American Standards Association's sectional committee B46—a committee concerned with surface texture.

HARVEY E. MARTIN, contributor of "The GM Portager: A New Concept in Piggyback Rail Car Design," is new product development engineer at General Motors Diesel Limited, London, Ontario. He is responsible for the engineer-

ing development work on new products. New Product Development comprises an engineering group at GM Diesel Ltd. which is separate from engineering activities devoted to products presently in production.

Mr. Martin joined GM Diesel Ltd. in 1951 as a project engineer. Two years later he was promoted to supervisor of project engineering. In 1955 he became chief project engineer. He assumed his present position in 1960. Included among his previous major projects was developmental work on a line of Diesel-hydraulic locomotives.

Mr. Martin received a bachelor of science degree from Queen's University, Kingston, Ontario, in 1949. He then worked as an engineer for Canadian General Electric Company before joining General Motors. He is a member of the Engineering Institute of Canada and is a registered member of the Association of Professional Engineers of the Province of Ontario.



DOONAN D. McGRAW, contributor of "Determining the Inventor and Related Matters," and coordinator of this issue's "Notes About Inventions and Inventors," is a patent attorney in the General Motors Patent Section, Detroit Office. His current work includes the preparation of patent applications and their prosecution before the U.S. Patent Office. He also negotiates patent licenses and contracts, and investigates proposed production items for possible patent infringement. His work concerns automotive engines, the vehicle as an overall unit, frames, road speed control systems, vehicle power package units and drive lines, and fasteners.

Mr. McGraw joined GM in 1948 as an experimental engineer with Allison Division after receiving the B.M.E. degree from the Georgia Institute of Technology. In 1951 he was recalled to active duty in the U.S. Navy. After his release in 1953, he joined the Washington Office of the Patent Section as a patent searcher. He received the LL.B. degree in 1956 from the George Washington University Law School and was transferred to his present position the same year.

Mr. McGraw is a registered patent attorney and is admitted to the Bar in the District of Columbia and Michigan. He also is a member of the Michigan and American Bar Associations, Michigan Patent Law Association, and Phi Alpha Delta law fraternity.

THEODORE L. ROSEBROCK, co-contributor of "Analyzing the Exhaust Velocity Requirements for Electrically Powered Rockets," is section chief, Advanced Propulsion Devices Section at Allison Division. This Section is part of the Research Activity, Engineering Research Department. Mr. Rosebrock supervises research activities in ion and plasma devices for space flight application. Such activities include the design and testing of experimental electrojets, theoretical and experimental research in magnetofluiddynamics and ion charge neutralization, and space mission analysis.

Mr. Rosebrock joined Allison in 1940 as an inspector in the Receiving Inspection Department. Subsequent promotions included experimental technician, experimental test engineer, superintendent of turbo-jet test projects, and group project engineering—propulsion systems research. He assumed his present position in 1957. Included in his past major projects has been the supervision over experimental test activities on gas turbine engines, compressors, combustion systems, and controls.

Mr. Rosebrock was granted the B.S. in aeronautical engineering degree from Purdue University in 1948. He is a member of the Institute of AeroSpace Sciences.

RICHARD W. THOMAS, contributor of "Ultrasonic Cleaning: Its Theory and Some Factors Affecting its Application," is a metallurgist in the Metallurgical Laboratory at Diesel Equipment Division.

Mr. Thomas joined Diesel Equipment in 1955. Some of his previous major projects have included investigations of braze alloy compositions for joining cast iron to steel and a study of hard chromium plating. His present projects, in addition to those dealing with the application of ultrasonic cleaning, include studies on the improvement of brazing processes.

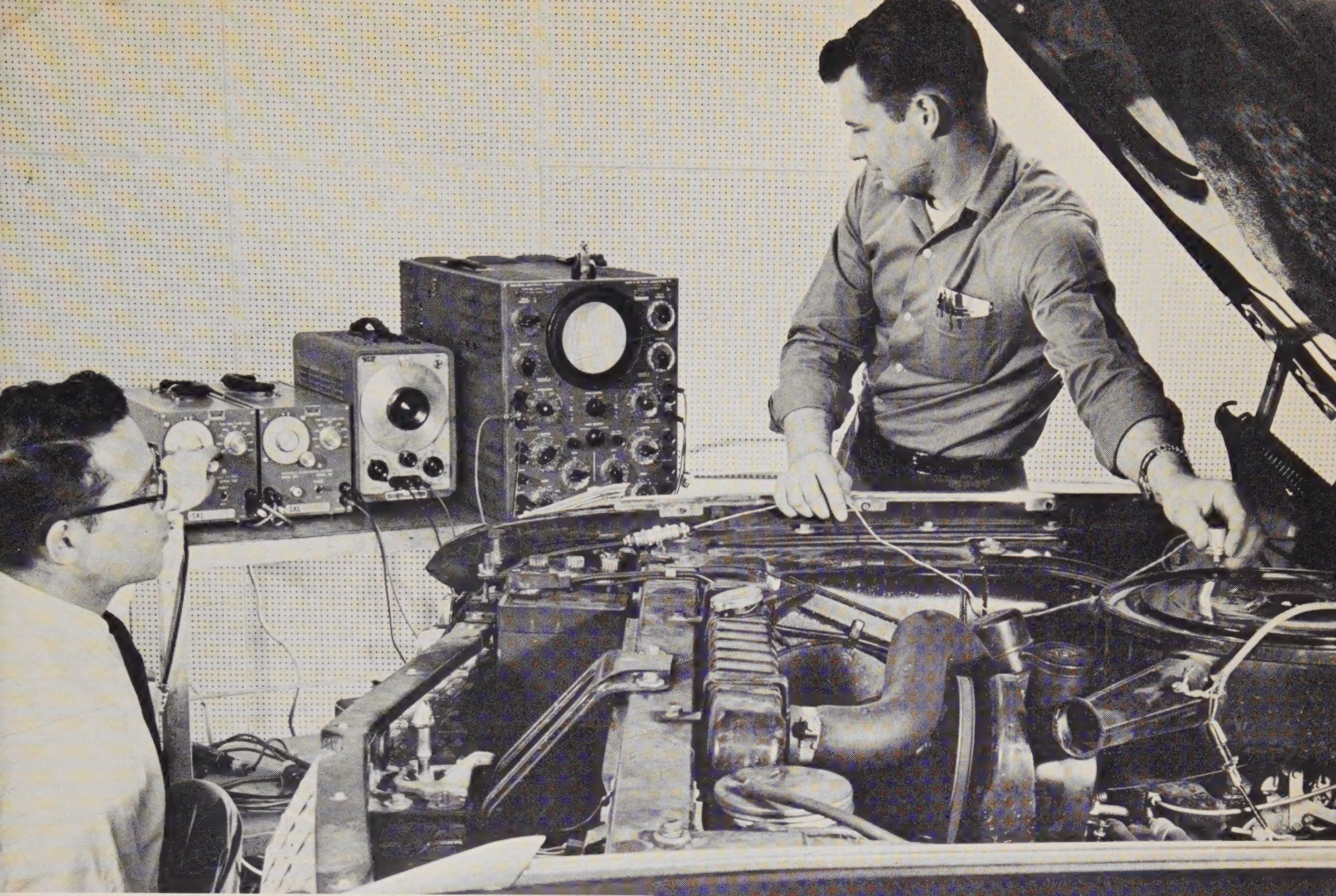
Mr. Thomas was granted a B.S. degree from the College of William and Mary in 1943. He is a member of the American Association for the Advancement of Science (Fellow), the Michigan Academy of Science, and the American Society for Metals. Prior to joining General Motors, he was chief chemist for the Military Division of Continental Motors.

HAROLD J. WARNER, Jr. contributor of "Applying Instrumentation to the Evaluation of Noise in Automotive Accessory Motors," is an experimental engineer in charge of the Noise and Vibration Section of the Electrical Laboratory of Delco Appliance Division. He is responsible for the development and utilization of methods of noise and vibration measurement and evaluation. Typical work includes the development of balancing specifications required to control vibrations and the development of standards for the selection of motor brush materials.

Mr. Warner received the B.M.E. degree from Clarkson College of Technology in 1953. He joined Delco Appliance the same year as an engineer-in-training. In 1954 Mr. Warner was transferred to the Product Engineering Laboratory and assigned to the Electrical Section to investigate noise and vibration problems. In 1957 he was promoted to senior tester on all noise and vibration problems. He was promoted to his present position in 1958.

The technical affiliations of Mr. Warner include the American Society of Mechanical Engineers and the American Society for Quality Control.

Contributors' backgrounds vary greatly in detail but each has achieved a technical responsibility in the field in which he writes.



ENGINEERING

ASSIGNMENT IN GM

Noise and vibration in automobiles continue to be challenging problems facing engineers, since the comfortable and reliable operation of a car depends in great measure upon the lack of vibration.

To help produce cars that are as free from noise and vibration as engineering and manufacturing methods permit, Cadillac Motor Car Division has a Noise and Vibration Committee consisting of 12 engineers, each at senior project engineer level or higher, representing all of the Division's design groups. Each member is assigned to special investigations in his particular field. This committee meets weekly to give progress reports and to coordinate plans for current noise and vibration studies.

The application of proper instrumentation is essential in noise and vibration studies, since engineers learned years ago that intuition or cut-and-try methods applied to these studies were inadequate.

In the photograph above, William Armstrong (left), senior project engineer in charge of vibration, Engineering Laboratory, and a member of the Committee, adjusts electronic filter equipment while Earl Fake, special tester, Engineering Laboratory, uses a vibration probe to locate the source of noise producing vibrations among engine accessory components. Other equipment used in this investigation includes an oscillator and an oscilloscope.

When the troublesome resonant component is identified, it is altered to reduce or eliminate the noise and vibration. These modifications are made by applying such methods as changing the stiffness of the component, thus shifting the vibration frequency; isolating the component from the source of vibration; or using insulating materials, such as rubber mounting, to absorb the vibration.

Mr. Armstrong joined Cadillac in 1950 as a junior engineer after his graduation from Iowa State University with a B.S.M.E. degree. He was promoted to project engineer in 1953 and to foreman in charge of vehicle tests in 1955. He assumed his present position in 1958.

Mr. Fake joined Cadillac in 1953 as a mechanic after attending the State University of New York at Morrisville where he was awarded a certificate in automotive technology. After a two-year military leave of absence, he rejoined Cadillac as a laboratory technician. He was promoted to his present position in 1956. He is currently studying towards a B.S.M.E. degree at Wayne State University.

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